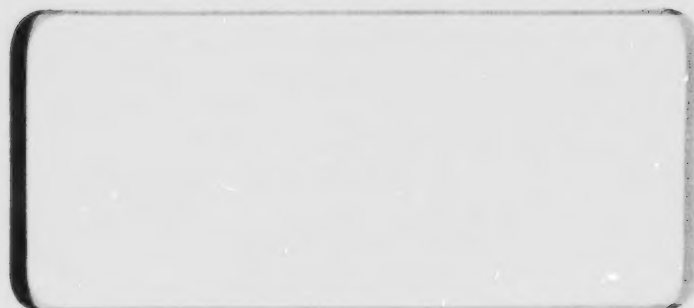




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TANK CAR THERMAL RESPONSE ANALYSIS:
VOLUME 4
POOL BOILING HEAT TRANSFER: THE EFFECTS
OF ENHANCED SURFACES AND INCLINATION

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17. Abstract An examination of pool boiling heat transfer in Freon-11 with different tank surfaces has been carried out. Surfaces of copper, mild steel, Union Carbide High Heat Flux and two UNB manufactured enhanced surfaces were investigated. The study focused on the development of heat transfer enhancing devices and the experimental evaluation of their performance. An investigation of the effects of surface orientation in both nucleate and film boiling regimes was carried out. An experimental apparatus was developed to facilitate the replacement of the test section with another and enable data to be obtained at any surface orientation in the natural convection, nucleate and film boiling regimes. Photographic studies of the cross sections of the prepared surfaces and surface roughness tests were conducted in an attempt to explain the bubble nucleation mechanism and film boiling results for each surface. It is concluded that surface characteristics, such as pore size, shape and distribution, are important factors in determining the heat transfer characteristics. Surface orientation is important in nucleate boiling for the region of the inclination angle of 165° to 180° (facing downward). This effect is also important in film boiling for the facing downward case, where the superheat required is much higher as compared to the other orientation. Finally, swelling height is found to be a strong function of heat flux and initial liquid filling in the tank.			
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17. Résumé Nous avons étudié le transfert de chaleur dans le fréon-11 en ébullition en convection naturelle dans des réservoirs à surfaces de types différents. Nous avons également étudié des surfaces de cuivre, d'acier doux, à flux thermique élevé fabriquées par Union Carbide et deux surfaces spéciales fabriquées par l'Université du Nouveau-Brunswick. L'étude a porté sur la mise au point de dispositifs visant à accroître le transfert de chaleur et sur l'évaluation expérimentale de leur rendement. Nous avons aussi examiné l'effet de l'orientation de la surface en régimes d'ébullition pelliculaire et d'ébullition germinée. L'installation expérimentale a été mise au point pour faciliter le remplacement de la section de test par une autre et obtenir des données à toutes les orientations de la surface, en régimes d'ébullition en convection naturelle, d'ébullition germinée et d'ébullition pelliculaire. Des études photographiques de coupes de surfaces et des mesures de rugosité ont été effectuées pour tenter d'expliquer les mécanismes de nucléation en bulles et d'ébullition pelliculaire pour chaque surface. Les résultats démontrent que les caractéristiques d'une surface, telles que le diamètre, la forme et la distribution des pores, sont des facteurs importants pour les transferts de chaleur. L'orientation de la surface est importante en régime d'ébullition germinée lorsque l'angle d'inclinaison se trouve entre 165° et 180° (face vers le bas). Ce facteur est également important en régime d'ébullition pelliculaire avec face vers le bas, la chaleur de surchauffe nécessaire étant alors beaucoup plus élevée que pour les autres orientations. Finalement, la hauteur de gonflement dépend fortement du flux thermique et du niveau de liquide initial dans le réservoir.			
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NOMENCLATURE

<u>Symbol</u>	<u>Description</u>
A, B, C, D	constants
C_{p_l}	specific heat
C_{sf}	leading constant in Rohsenow equation
g	acceleration of gravity (m/sec^2)
h	heat transfer coefficient ($\text{W/m}^2\text{°C}$)
h_{fg}	latent heat of vaporization (J/kg)
Gr	Grashof number
Gr_c	critical Grashof number
k	thermal conductivity ($\text{W/m}^{\circ}\text{C}$)
Nu	Nusselt number
P	pressure
Pr	Prandtl number
q/A	heat flux (W/m^2 or kW/m^2)
$(q/A)_{\min}$	minimum heat flux (W/m^2 or kW/m^2)
r, s	exponents in Rohsenow equation
T_{sat}	saturation temperature ($^{\circ}\text{C}$)
T_w	wall temperature ($^{\circ}\text{C}$)
ΔT	superheat, $T_w - T_{\text{sat}}$ ($^{\circ}\text{C}$)
$\Delta T_{\min}$	minimum temperature difference ($^{\circ}\text{C}$)
t	time (sec)
V	velocity (m/sec)
α, θ	inclination angle (radian or degree)

<u>Symbol</u>	<u>Description</u>
μ	viscosity (Pa·sec)
σ	surface tension (N/m)
ρ	density (kg/m ³)
ℓ, l	liquid
v	vapour

CHAPTER 1

INTRODUCTION

1.1 Introduction

Railroad tank cars transporting such hazardous fluids as propane, butadiene, ammonia, ethylene, vinyl chloride and chlorine have been involved in a rising number of accidents. These accidents have caused death, serious injury, extensive property damage, and disruptive major evacuations.

When a typical tank car is exposed to fire (Fig. 1.1), heat is conducted through the tank wall and dissipated into the bulk liquid. The change of phase of the tank contents, (liquid to vapor) is governed by the Clapeyron equation.

$$\left(\frac{dP}{dT}\right)_{\text{sat}} = \frac{h_{fg}}{T_{\text{sat}} v_{fg}} \quad (1.1)$$

where h_{fg} and v_{fg} are the latent heat of vaporization and difference of the specific volumes of the two phases respectively.

Until this equation is satisfied only single phase convection exists but as the heat flux increases, eqn. (1.1) is satisfied and two phase boiling takes place inside the tank. Subsequently bubbles are formed on the tank wall

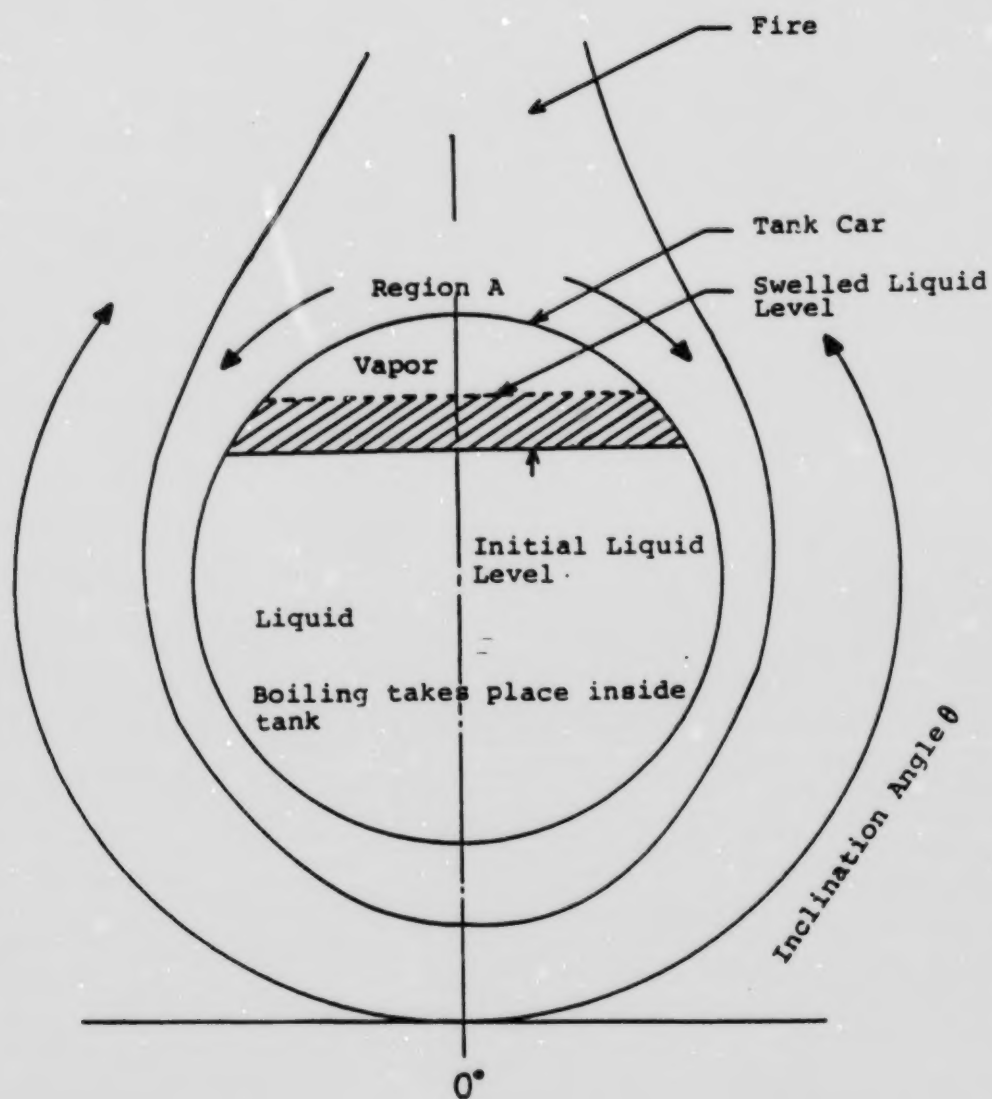


Fig.1.1 Typical situation of railroad tank car in case of fire condition.

in the process of subcooled boiling and eventually the bubbles rise to the liquid surface if the heat flux at the wall is great enough. Due to this bubble generation there is a significant change in liquid level as shown in Fig. 1.1.

Tank cars normally utilize large horizontal cylinders and hence the interior fluid sees a variety of surface inclinations and heat fluxes.

In order to make the shipping of hazardous industrial fluids safer, a complete understanding of the actual phenomena inside tank cars under external fire condition is necessary. As part of this work studies were carried out on i) pool boiling phenomena and heat transfer performance as a function of the surface orientation, ii) development of the heat transfer enhancing device such as internal protection system (Explosafe) and metal coated porous surfaces, and iii) two phase fluid swelling phenomena so as to know the level change with boiling.

1.2 Study Objectives

In order to understand the above phenomena the present work was initiated with the following objectives:

- 1) an experimental determination and comparison of the performances of the heat transfer enhancing devices which include Explosafe and various surfaces with a porous coating;

- 2) an experimental investigation of the effect of the surface orientation on pool boiling; and
- 3) a theoretical prediction for the swelling height and comparison with experimental data.

In the following chapter the relevant literature on pool boiling phenomena including enhanced surface effects, the effect of orientation and two phase fluid swelling is reviewed. The experimental design, setup and test procedures are discussed in Chapter 3. The theoretical analysis for swelling is given in Chapter 4. The results of the present study and their discussion are presented in Chapter 5. Conclusions from this work and suggestions for future work are presented in the last chapter.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

Pool boiling is the boiling mode that occurs when a heater is submerged in a pool of initially stagnant liquid. Pool boiling represents the traditional starting point for a discussion of heat transfer in boiling systems. Due to extensive research efforts over the past thirty years, the basic mechanisms of pool boiling are relatively well understood.

Cole [1] has critically discussed the most important publications pertaining to the theory of nucleation in liquids in a recent survey article. However, it is still not possible to predict the heat transfer characteristics for this simplest of boiling systems with the precision associated with single-phase systems. This is due largely to the inherent difficulty of quantitatively describing all the important and interacting surface and fluid characteristics.

There are four different regimes in the boiling curve. As shown in Fig. 2.1, the boiling characteristics are generally represented as a log-log plot of heat flux versus wall superheat.

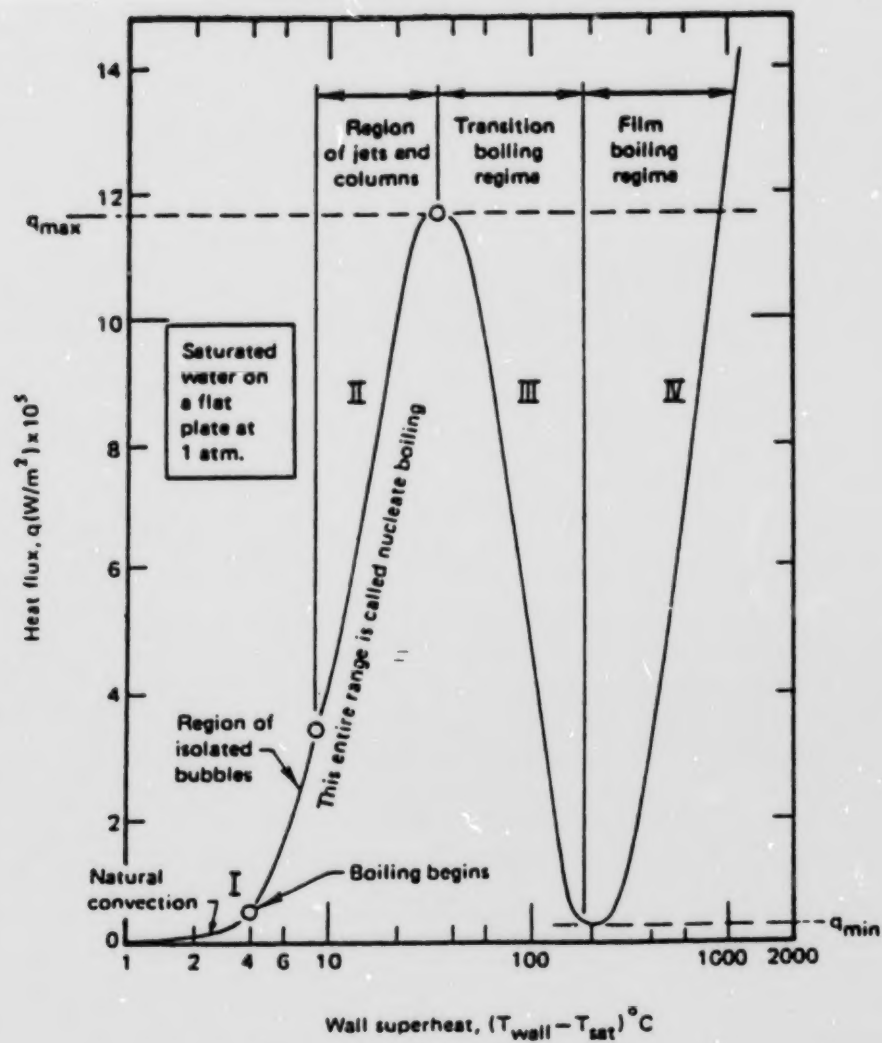


Fig.2.1 Typical boiling curve and regimes of boiling (from [2]).

In region I free convection currents are responsible for motion of the fluid near the surface. In this region the liquid near the heated surface is superheated slightly and it does not subsequently evaporate.

In region II bubbles begin to form on the surface and are dissipated in the liquid after breaking away from the surface. This region indicates onset of nucleate boiling. As the temperature excess is increased further, bubbles form more rapidly and rise to the surface of the liquid, where they are dissipated. The relation between the heat transfer rate and the temperature of the heating surface in nucleate boiling depends on the thermodynamic and transport properties of both phases of the fluid, on the nature of the heating surface, its orientation in the field of gravity and subcooling. Eventually, bubbles are formed so rapidly that they blanket the heating surface and prevent the inflow of fresh liquid from wetting the surface. At this point the bubbles coalesce and form a vapor film before they can reach the liquid and effect the boiling process. The thermal resistance of this film causes a reduction in heat flux (region III). This region represents a transition from nucleate boiling to film boiling and is unstable. Region IV represents the stable film boiling where the surface temperatures required to maintain stable film boiling are high and the heat transfer occurs through the stable vapor film covering the surface

and separating it from the liquid.

A detailed literature review regarding pool boiling heat transfer will not be presented here but for more information excellent reviews are given by Rohsenow [3], Bergles et al. [4], Clark [5] and Collier [6]. Literature regarding the effects of enhanced surfaces, surface orientation and two phase swelling on boiling heat transfer will, however, be reviewed due to their importance to this study's objective.

2.2 Effect of Surface Orientation

In 1953, Rich [7] showed that heat transfer from inclined plates in natural convection could be predicted by vertical plate formulas if the component of gravity vector along the surface of the plate was used in the calculation of the Grashof number. Thus, the heat transfer rate decreases as $(\cos \alpha)^{1/4}$, where α is the angle measured from the vertical as shown in Fig. 2.2.

Subsequent studies [8, 9] have shown that Rich's result is substantially correct for the lower surface of a heated plate. For the upper surface of a heated plate, the boundary layer becomes unstable and separates at a lower value of Grashof number. Experimental observations of such instability have been given by Fujii and Imura [8] and Vliet [9]. Fujii and Imura suggested the following for different cases based upon experiments in water.

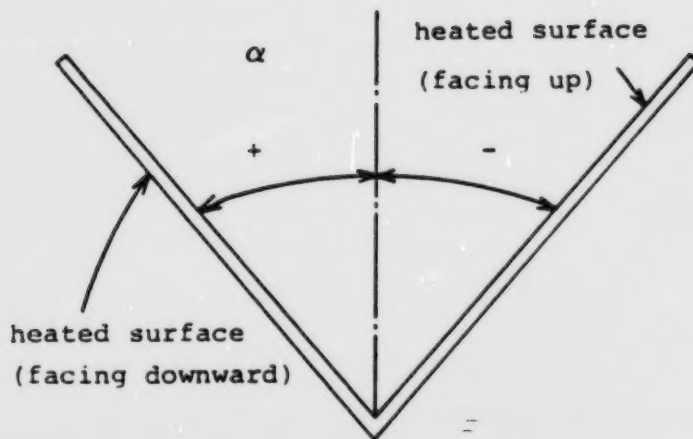


Fig.2.2 Coordinate system for inclined plates
(from[8]).

Note: The inclination angle α in the Fig. 2.2 is different from θ which represents the inclination angle in the present study (Fig. 1.1). The relation between them is

$$0^\circ < \alpha < 90^\circ \rightarrow 90^\circ < \theta < 180^\circ$$

$$-90^\circ < \alpha < 0^\circ \rightarrow 0^\circ < \theta < 90^\circ$$

1. For the inclined plate facing downward:

$$\overline{Nu} = 0.56(GrPr \cos \alpha)^{1/4} \quad \alpha < 88^\circ, \quad 10^5 < GrPr \cos \alpha < 10^{11} \quad (2.1)$$

2. For the almost horizontal plate facing downward:

$$\overline{Nu} = 0.58(GrPr)^{1/5} \quad 88^\circ \leq \alpha \leq 90^\circ, \quad 10^6 < GrPr < 10^{11} \quad (2.2)$$

3. For the horizontal plate facing upward:

$$\overline{Nu} = 0.16(GrPr)^{1/3} \quad \alpha = -90^\circ, \quad 10^6 < GrPr < 10^{11} \quad (2.3)$$

4. For the inclined plate facing upward (unstable case):

$$\overline{Nu} = 0.13[(GrPr)^{1/3} - (Gr_c Pr)^{1/3}] + 0.56(Gr_c Pr \cos \alpha)^{1/4} \quad (2.4)$$

$$-15 \leq \alpha \leq -75, \quad 10^5 < GrPr \cos \alpha < 10^{11}$$

All properties except β are evaluated at a reference temperature T_e defined by:

$$T_e = T_w - 0.25(T_w - T_\infty) \quad (2.5)$$

where T_w is the mean wall temperature and T_∞ is the free stream temperature. The quantity Gr_c is a critical Grashof number dependent upon the inclination angle and it indicates when the Nusselt number starts to deviate from the laminar relation given by eqn. (2.1).

The effect of the orientation of the heating surface in nucleate boiling has been reported by Githinji and Saberski [10]. They made comparisons of boiling heat

transfer from a strip in isopropyl alcohol for three different surface orientations - facing up, facing downward and vertical (Fig. 2.3).

Curves A and B exhibit the characteristics typical for nucleate boiling heat transfer, that is the sharp increase of heat transfer rate with surface temperature after the onset of nucleation and the existence of a burnout point. Curve C, however, differs significantly from the other two. Curve C does not show any strong changes in slope and resembles more nearly that for ordinary free convection without phase change. Nevertheless, a burnout point is also observed. The heat flux rate at the burnout point for the strip facing downward is, however, several times lower than that for the strip facing upward.

Li-Ting Chen [11] performed similar experiments and obtained data at inclination angles from 0° to 180° in R-11. He correlated the data using Rohsenow's equation given by eqn. (2.6) and concluded that the orientation of the surface has no effect on the exponent 'r' and the constant ' C_{sf} ' decreases as the inclination angle, θ , is increased from 0° to about 150° .

$$\frac{C_p \Delta T}{h_{fg}} = C_{sf} \left(\frac{q/A}{\mu_l h_{fg}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right)^r \left(\frac{C_l \mu_l}{K_l} \right)^s \quad (2.6)$$

To the author's knowledge, there has been no reported work for film boiling from inclined surfaces.

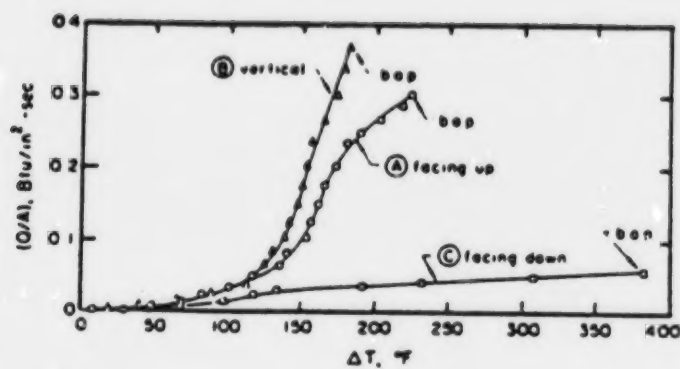


Fig.2.3 Effect of orientation on nucleate boiling(from[10]).

2.3 Boiling Phenomena From Enhanced Surfaces

Industrial demands for more efficient boilers and evaporators have spurred the development of methods of enhancing boiling heat transfer coefficients or critical heat fluxes. Of greatest interest are the special surfaces which promote nucleate boiling at low temperature differences. As outlined in reference [12], one of the categories of a structured boiling surface is metal coated surfaces, whereby an irregular matrix of potential nucleation sites is produced by such methods as poor welding, sintering or brazing of particles, electrolytic decomposition, or the bonding of particles by plating or metallic coating of a foam substrate.

The general objective of these surface treatments is to provide nucleation sites which will be stable, that is, trap vapor so that initial bubble formation can occur at moderate wall superheat. These sites, furthermore, promote high rates of vaporization so that the wall superheat required for developed nucleate boiling is low compared to that required for boiling from surfaces with natural cavities.

Fig. 2.4 shows one model for a porous matrix coated on a surface. If the matrix material is poorly wetted by the liquid, vapor or vapor plus noncondensable gas will be retained in the interstitial space when the temperature is reduced below the saturation temperature. Alternatively, in the more usual wetting situation, re-entrant cavities are required to permit stable dropwise

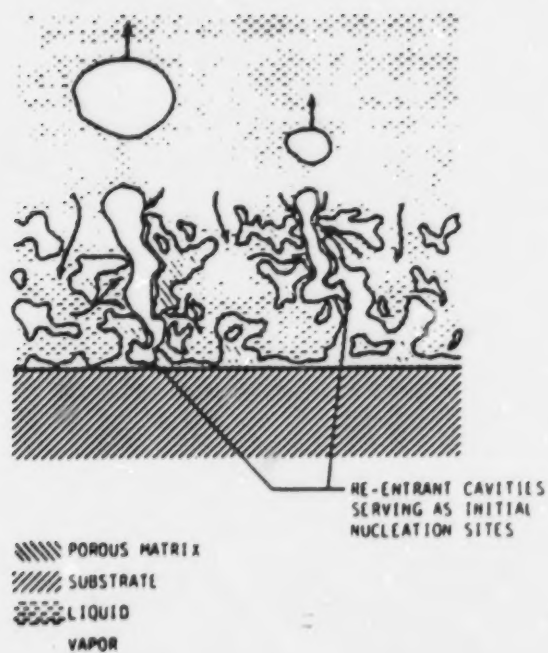


Fig.2.4 Conceptual model of boiling
in a porous matrix(from[15]).

formation so that the cavities are not filled with sub-cooled liquid. In any event, the matrix increases the probability that nucleation sites are available which will remain active for repeated cycles of heating and cooling. Furthermore, there is a greater probability that the cavities will have larger effective mouth radii than natural surface cavities, thereby insuring incipient boiling at lower wall superheat.

Several previous investigators [13, 14] postulated the model of nucleation in a porous matrix. Good reviews regarding their models are given in [12, 15] and will not be repeated here.

2.4 Effect of Swelling

There will be a boiling on the tank wall as the heat flux increases resulting in vigorous bubble formation. The bubbles released at the surface have a finite residence time in the liquid column above the heated surface. This 'bubble holdup' increases the level of the two-phase boiling system as seen in Fig. 1.1. It is important to know the level change in order to determine the free space available for disengagement of the entrapped droplets and to be able to determine the physical properties such as void fraction, which in essence, affects the boiling process inside the tank. Singh and Gupta [16] have analyzed these

phenomena for a constant pressure situation, in which boiling is assumed to originate uniformly over the heater surface immersed in the fluid. The bubbles formed move through the liquid and burst up at the top of the liquid surface. The residence time of the bubbles in the liquid causes an increase of the total volume and hence the level of the boiling liquid or the two phase system increases in the tank. Therefore, to know the swelled height due to bubble holdup will determine the free space available and liquid carryover and hence the capacity of the remaining liquid to carry away the heat in the form of the latent heat of vaporization.

CHAPTER 3

EXPERIMENTS

3.1 Apparatus

The experimental apparatus is composed of five parts: an electric heater, test specimens (copper and mild steel surfaces, three kinds of enhanced surfaces (UNB surface #1, #2, and Union Carbide High Heat Flux surface)), T-vessel and its equipment, an electric circuit panel, and thermocouples along with a HP-85 computer and data acquisition system (3456 A and 3497 A digital voltmeters). Details of this apparatus are described below.

Electric heater: To achieve a uniform temperature field on the surface, a coil heater was used. Fig. 3.1 illustrates the detailed cross sectional view of the electric heater. Spiral grooves for the coil were machined on the insulation block made of machinable ceramic (Lava; $k = 1.3 \text{ w/m}^{\circ}\text{k}$) which limits the heat transfer to one side of the heating plate. Fig. 3.2 shows the top and side views of the heating element. After the coil, which is made of nichrome wire (16 gauge, 3.3 m), was put on the grooves, spaces between the grooves and coils and inside coils were filled with cement (Al_2O_3) for better heat transfer. Four holes were drilled radially

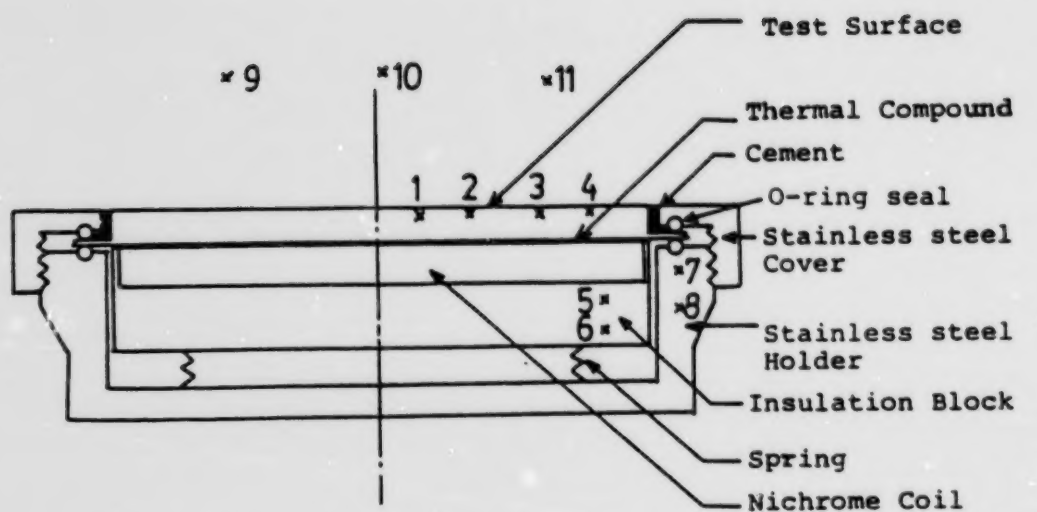


Fig.3.1 Detailed cross sectional view of the electric heater(actual size).

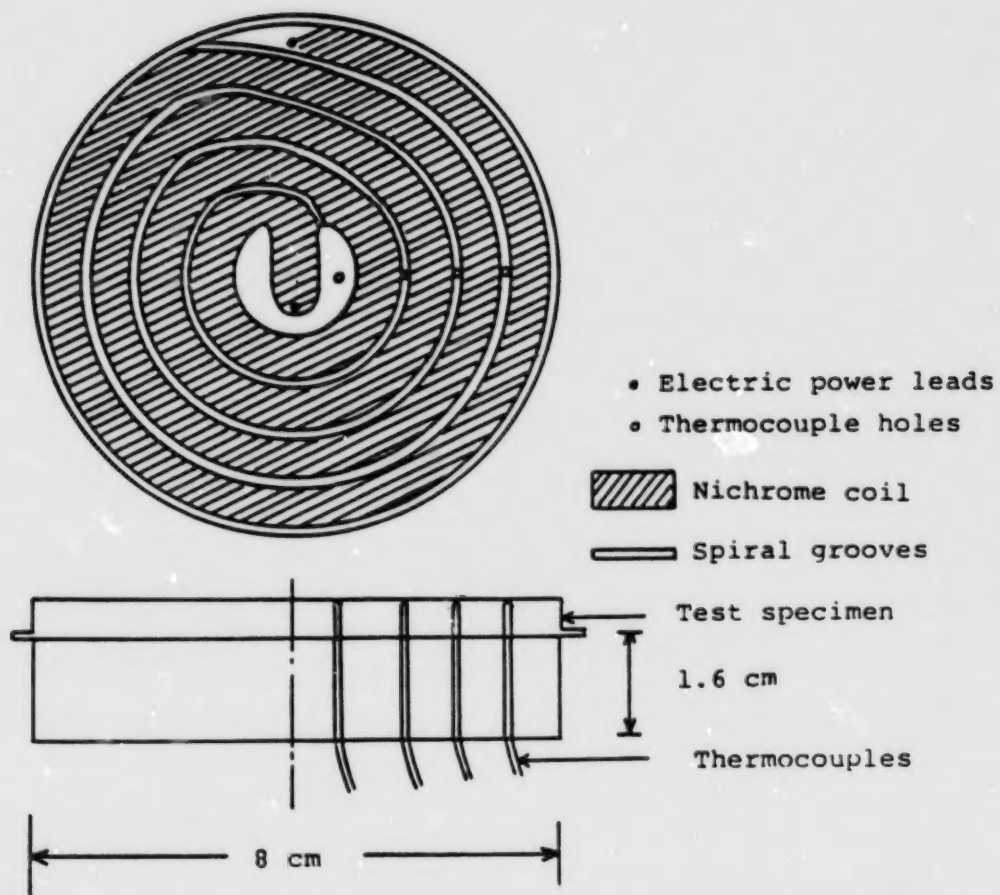


Fig.3.2 Top and side views of the heating element.

on the spiral grooves for inserting the thermocouples in the surface (Fig. 3.2). Next, this encapsulated heating element was ground for good contact with the test surface as shown in Fig. 3.3. A very thin thermally conductive compound (Wakefield Engineering, part no. 120-8) was applied between the test surface and the ground heating element for good thermal contact. A threaded stainless steel container with an easily removable cover was machined to facilitate the replacement of the test section with another and to hold the encapsulated heating element and test section. To limit the heat loss through the edges of the test section, the space between the side of the test section and stainless steel cover was filled with an insulation cement as seen in Fig. 3.1. Two 1/8" diameter o-ring seals were used on the grooves in the stainless steel holder and cover to prevent liquid penetration into the container. Three 0.9 cm diameter, 0.6 cm high springs were placed vertically between the insulation block and stainless steel holder to maintain a good contact between the ground heating element and test section.

For tests with Explosafe the material was cut in circular shape by an electric cutter and put on the surface. To maintain good contact, six 3/16" diameter threaded studs with accompanying bolts were used to secure wires over the surface of the Explosafe as shown in Fig. 3.4.

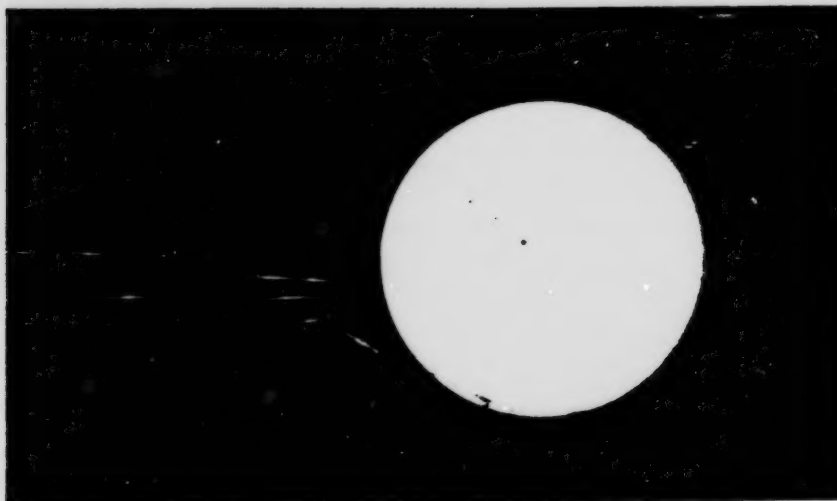


Fig.3.3 Encapsulated heating element.

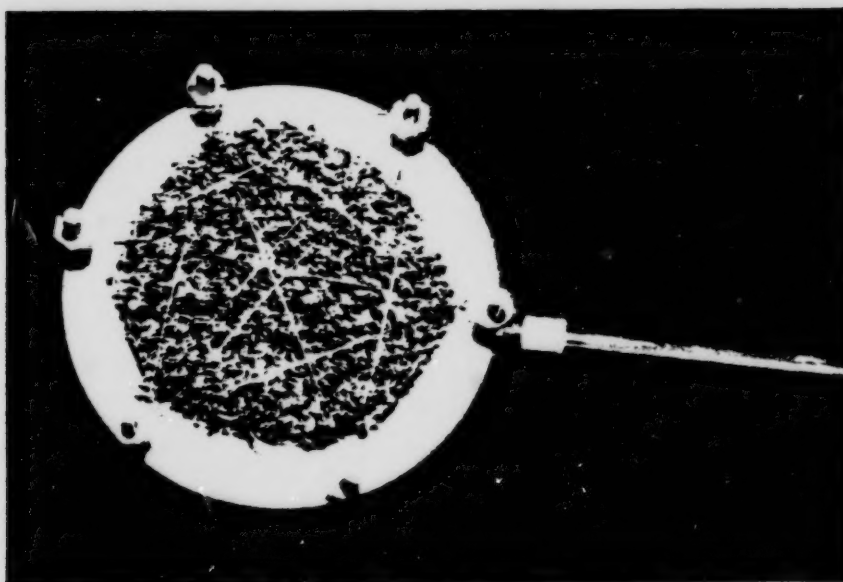


Fig.3.4 Heater assembly with Explosafe.

By adjusting the tension of the wire, good contact could be obtained. Fig. 3.4 illustrates the fully assembled heater with an Explosafe cover.

Test specimens: Two ordinary flat machined and finished surfaces (copper and mild steel) and three enhanced surfaces (UNB #1, #2 and UC) were prepared. Since the bubble diameter as it just leaves the surface in nucleate boiling is approximately 0.2 cm [17], there may be significant edge effects with surfaces of only 0.3 cm width [10] and 2.5 cm width [11]. In order to model large systems with reasonable confidence, a careful study with large surface area must be undertaken. To meet this purpose a 7.8 cm diameter circular test section was used here. As seen in Figs. 3.1 and 3.2, all specimens were grooved along the edges to fit the stainless steel holder with o-ring seals so that liquid could not penetrate into the heater. Four thermocouple holes were made just underneath the surface (Fig. 3.2).

The plain surfaces were polished with a coarse emery paper. Porous metal coating on the mild steel surface was done by flame spraying. A detailed procedure for the UNB surface #2 is given in Appendix A.

T-vessel and its equipment: As seen in Figs. 3.5 and 3.6, a pyrex glass vessel with T-section (43 cm tall, 15 cm diameter at both ends and 10 cm diameter at T-section)

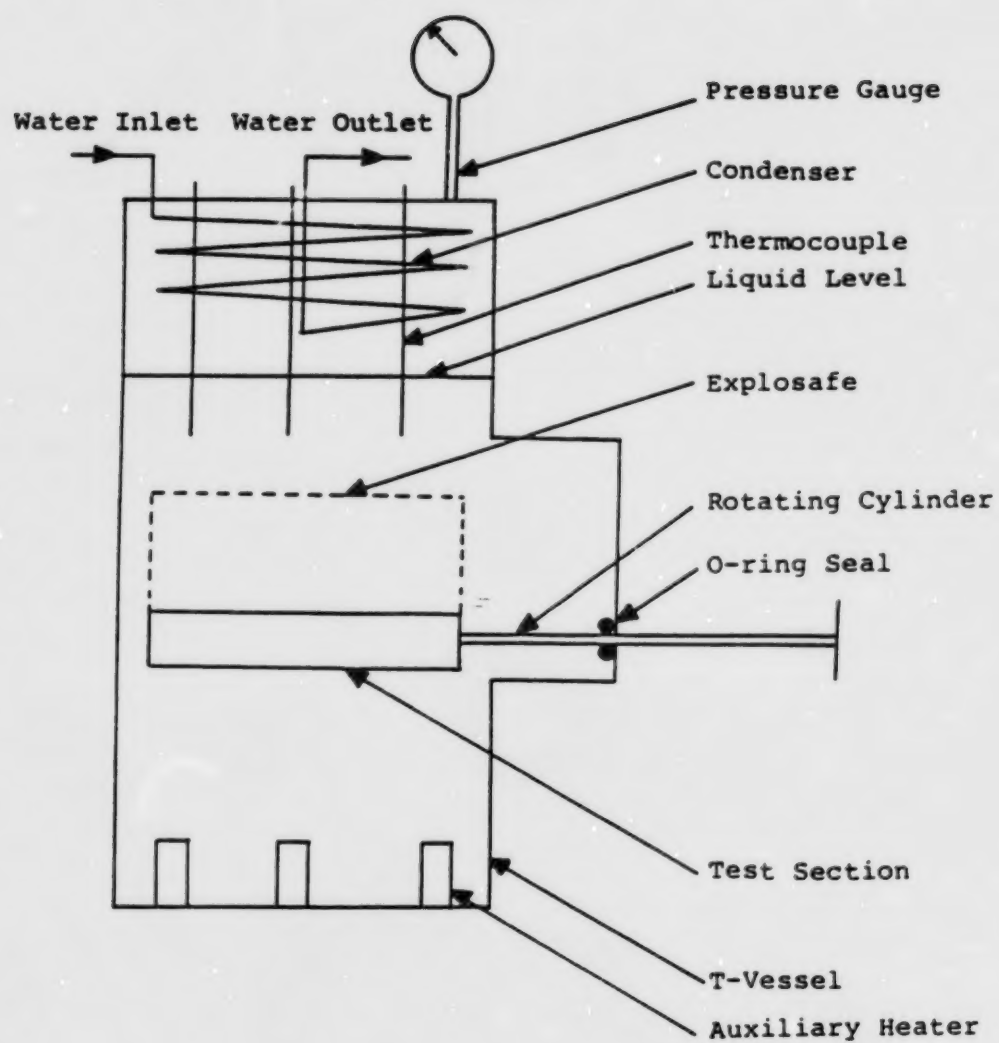


Fig.3.5 Schematic view of the apparatus.

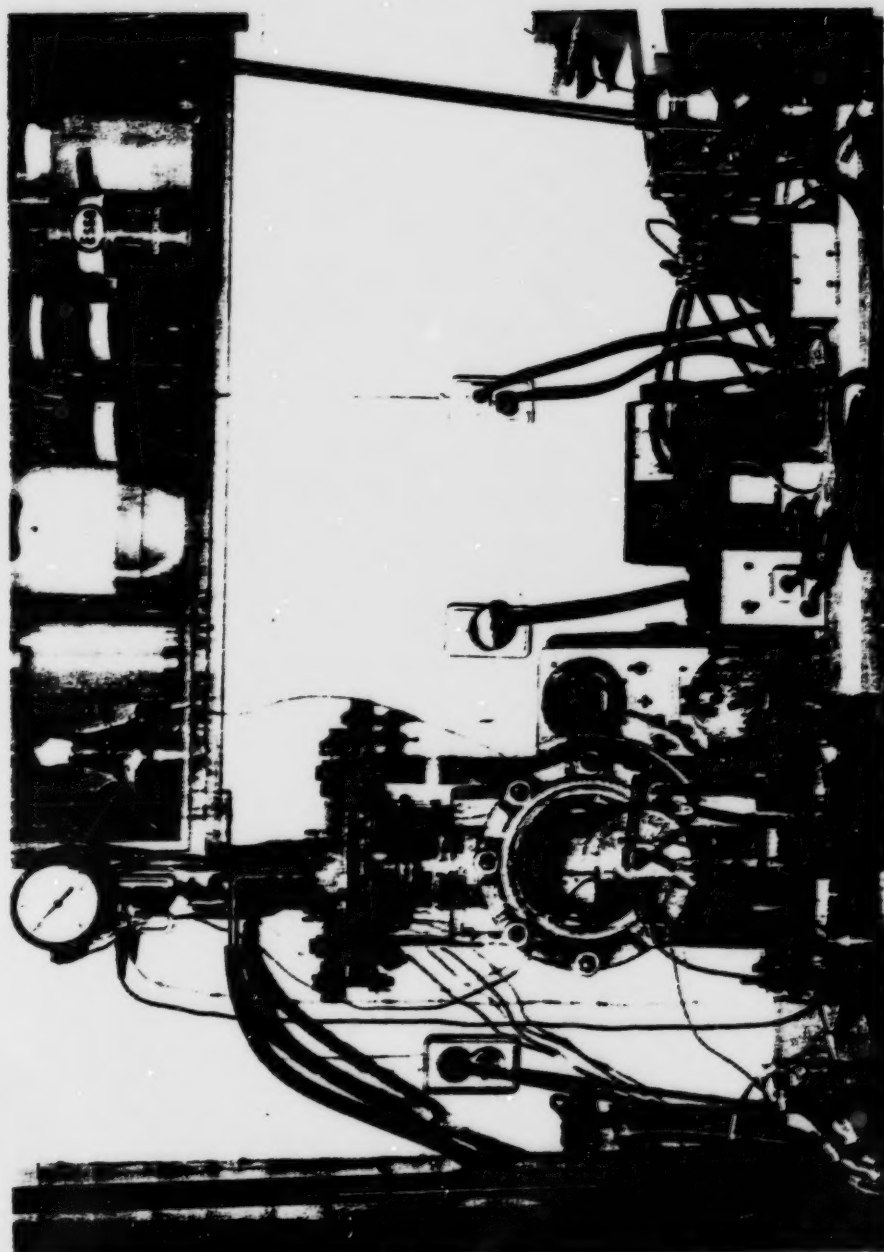


Fig.3.6 Photographic view of the apparatus.

was used as the pool. Two plexiglass ends (28 cm diameter, 2.5 cm thick) and an aluminum plate (28 cm diameter, 1.2 cm thick) served as covers for the top, side and bottom of the T-vessel respectively. Four holes were made on the top plexiglass to mount a pressure gauge and three thermocouple holders. Three holes were prepared on the bottom aluminum plate to install three auxiliary heaters. One hole was drilled on the side plexiglass for the rotating cylinder ($3/8$ " diameter, 10" long stainless steel pipe containing power and thermocouple leads).

To preheat the liquid to a desired temperature and maintain the system pressure especially at low heat flux, three auxiliary heaters (Dalton Electric Heating Co., 450 watts, 110 volts) in aluminum holders were installed on the lower fitting plate.

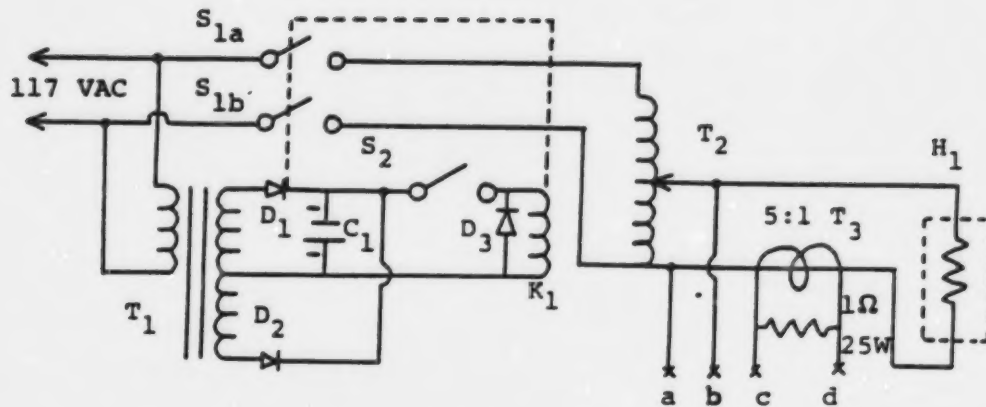
To maintain constant pressure inside the T-vessel during liquid boiling, a reflux condenser was utilized. The condenser is of a coil type and made of 1 cm outside diameter, 140 cm long copper tubing and suspended under the top plexiglass (Fig. 3.5). The required cooling water of the condenser was circulated inside the tube. To observe the pressure variation inside the pool a pressure gauge was installed on the top plexiglass (Fig. 3.5). A protractor and an indicator for surface orientation were fixed on the rotating cylinder. As seen in

Fig. 3.5, an o-ring seal was used to prevent liquid leakage at the intersection due to pressure.

Electric circuit: To control the power input into the electric heater and operate it, an electric circuit panel was utilized. As seen in Fig. 3.7, a standard resistor (H. Tinsley & Co. Ltd., Type 1659, 0.5Ω , 4.5 amps) was used in order to measure the true current through the nichrome coil. While the system undergoes high heat flux, the current through the coil far exceeds the limit for the resistor. To avoid the resistor's breaking down, a current transformer was used to reduce the current going into the resistor.

In case the heater experiences a transition from nucleate to film boiling at high heat flux ranges, there is an abrupt temperature increase on the surface and the heater may burn out unless prevented by proper means. To maintain heater integrity, a relay was used to shut down the power in case of sudden surface temperature jumps. The operation of the heater and measurement of the voltages across the heater and standard resistor were made possible by the use of a HP-85 computer and data acquisition system which will be described below.

Thermocouples and data acquisition system: As seen in Fig. 3.1, eight iron-constantan thermocouples (Thermo Electric, 30 gauge) were used to measure the temperatures



- $D_{1,2,3}$: 600V-1A Diode
 C_1 : 470 μ f/35V
 K_1 : 24 V Coil DPST Relay
 $S_{1a,b}$: DPST Relay 24 Volt DC Coil
 S_2 : Remote control (HP-3497 voltmeter)
 a,b : E_1 , Voltage through main heater
 c,d : E_2 , Voltage through standard resistor
 T_1 : 36VCT/117V
 T_2 : 12A/0-140V, Voltage transformer
 T_3 : 5:1, Current transformer
 H_1 : Test heater

Fig.3.7 Detailed electric circuit.

on the surface, insulation block and stainless steel holder. Several authors [18, 19] have performed the boiling tests, where they measured a temperature gradient and extrapolated it to obtain the surface temperatures and heat flux instead of measuring surface temperatures directly. As for their results, the accuracy of the measurements is completely dependent upon the thermal conductivity value of the material and the thermocouple placements. In the present experiments, direct surface temperature measurements were attempted. As shown in Fig. 3.2, spot welded thermocouples were inserted through the holes in the insulation block. Then, they were pasted with a high thermal conductivity cement (Teledyne Dental) and squeezed into the holes which are just underneath the test section. Another two thermocouples were cemented inside the insulation block to measure the heat loss through the bottom of the heater as seen in Fig. 3.1. The other two thermocouples were cemented inside the stainless steel holder to measure the heat loss through the sides of the heater. Detailed locations of the thermocouple no. 5 ~ 8 (Fig. 3.1) are given in Appendix C.

To measure the liquid temperatures three copper-constantan thermocouples (Thermo Electric, 30 gauge) were immersed in the pool through the thermocouple holders in the top plexiglass as shown in Fig. 3.5.

Monitoring of these eleven thermocouples and two AC voltages was done by the use of a HP-85 computer and data acquisition system (Fig. 3.8). The HP-85 computer adopts BASIC as the operational language with a 32 kilo bites memory. Software can be stored in either floppy disks or data cartridges. There is a fast I/O buffer available for transferring data from or to peripheral devices such as digital voltmeters. The voltmeter, HP 3497 A is a peripheral device with 100 input/output channels. The voltmeter is fully programmable remotely under the computer control via the HP Interface Bus. DC volts from thermocouple junctions are measured by this voltmeter. The basic function of the computer is to operate the mercury wetted relay switches and cause the voltmeter to read the voltage on input channels using A/D units. Resolution of the voltmeter is 1 μ V for the range 0.1 V. Measurement accuracy (% of reading, auto-zero on, 24 hours: $23^{\circ}\text{C} \pm 1^{\circ}\text{C}$) is 0.003% for the range 0.1 V.

The Integrating Digital Voltmeter, HP 3456 A is used to measure AC voltages. This voltmeter is capable of 330 readings/sec depending on the number of integration cycles. Ten integration cycles are selected. Resolution for AC RMS voltage is 100 μ V for the range 10 V. Measurement accuracy (% of reading, 24 hours: $23^{\circ}\text{C} \pm 1^{\circ}\text{C}$) is 0.53%.



Fig.3.8 HP-85 computer and data acquisition system.

3.2 Experimental Procedure

For a nucleate boiling test the experimental procedure for an individual run consisted of the following steps:

- 1) The heat transfer surface was cleaned before testing with acetone.
- 2) The liquid was changed and preheated to bring the system up to the desired pressure by the auxiliary heaters and subsequently degased.
- 3) Power to the main heater was initiated for the chosen orientation angle. At low heat flux, i.e., below 10 kw/m^2 (natural convection regime), the auxiliary heaters were turned on to maintain the system pressure. However, at high heat flux, i.e., above 10 kw/m^2 (boiling regime), those heaters were turned off and cold water was circulated through the condenser to ensure constant pressure inside the pool.
- 4) After the system reaches steady state condition, the data for the prespecified orientation was taken. The criteria for the steady state are quite strict. Within the period between the present and previous scans, which is approximately 12 seconds, i) temperatures from the thermocouple number 5 and 6 in the insulation block vary within the order of 0.02°C and ii) averaged surface and liquid temperatures, T_{surf} and T_{liq} , vary within the order of 0.02°C .

5) With the same heat flux, surface orientation was changed to the next desired inclination angle. Through all the experiments, the inclination angles at which data were taken with a given heat flux are: 0° (horizontal), 45° , 90° (vertical), 135° , 150° , 160° , 170° , and 180° (facing downward).

6) After data at specified angles was obtained, the power level was increased. The following items were measured at each data point; two voltages, four surface temperatures, three liquid temperatures, two temperatures in the insulation block, and two in the stainless steel holder. The power is increased just before the power limit of the heater is reached so as to generate a complete boiling curve. The heat flux range was approximately $1.5 - 170 \text{ kw/m}^2$ in nucleate boiling.

The experimental procedure for obtaining film boiling is quite different from that for nucleate boiling. The film boiling for the facing downward case was generated and then the rotating cylinder was turned to the desired surface orientation, maintaining a sufficient heat flux to sustain the film boiling by adjusting the voltage to the main heater. The advantage of this technique is that film boiling can be established directly at any surface orientation without having to go over the peak of the boiling curve, which might result in burning out the heater. The heat flux range in film boiling is $18 - 90 \text{ kw/m}^2$.

It generally took 20 ~ 30 minutes to obtain steady state condition after the power level was changed and 9 ~ 11 minutes was needed after the inclination angle was altered.

3.3 Instrumentation and Data Acquisition Procedures

The power supplied to the main heater was adjusted by a variable transformer (Type 3PN1010, output voltage range = 0 ~ 140 volts) and is monitored by a HP 3456 A digital voltmeter. The power supplied to the auxiliary heaters was controlled by another similar transformer.

Software was written (Appendix B) and stored on a data cartridge, which instructed the computer to command the 3497 A and 3456 A digital voltmeters to close a selected channel for thermocouple emf inputs and AC voltages respectively. A measurement flow diagram is also given in Appendix B with the list of principal variables.

After scanning the two AC voltages and eleven thermocouple emf values, the computer subsequently converts the emf to the actual temperature using a software compensation method with the accuracy of $\pm 0.1^{\circ}\text{C}$. The data acquisition system has its own reference isothermal block so that an ice bath, which usually serves as a reference junction in hardware compensation method, can be eliminated in the measurement circuit to facilitate the conversion of the emf

to temperature. At each scan, the four surface and three liquid temperatures are averaged as T_{surf} and T_{liq} respectively. To generate one data point on the boiling curve, usually 10 ~ 15 scans taken over a period of 3 ~ 5 minutes are used and the averaged T_{surf} and T_{liq} are computed. The standard deviations of these measurements are usually 0.01 ~ 0.03°C for nucleate boiling and 0.05 ~ 0.2°C for film boiling.

The superheat, ΔT is calculated by taking the difference between the averaged surface and liquid temperatures. The computer also calculates the total power input to the heater using the two averaged AC voltages across the main heater and standard resistor.

The heat flux associated with the boiling from a test section is subsequently obtained by subtracting the heat loss through the insulation block to the bottom and sides of the stainless steel holders determined from the temperature measurements (Appendix C). Heat losses related to natural convection, nucleate, and film boilings are approximately 14 ~ 15%, 10%, and 17 ~ 22% respectively. Heat losses with inclination do not change significantly (less than 1%) even though the inclination angle varies from 0° to 180°. The maximum error bound in the evaluated heat transfer coefficient is estimated to be less than 3.5%. The detailed heat loss calculation

method is given in Appendix C.

Fig. 3.9 illustrates a typical computer output data strip in which the real heat flux and temperature difference are used to represent a point on the boiling curve.

```

*****
TODAY IS FEB.2

INCLINATION ANGLE = 0

SURFACE TEMP. = 45.7526462813
SIGMA 1 = 1.88148877222E-2
LIQUID TEMP. = 43.8291460781
SIGMA 2 = 1.09027519462E-2
TEMP. DIFFER. = 1.9235002032
AVERAGE VOLT 1 = 5.048178
AVERAGE VOLT 2 = .1448468
POWER INPUT = 7.3121242913
LAVA TEMP. 1 = 50.0695017198
LAVA TEMP. 2 = 50.0978888672
STEEL TEMP.1 = 44.283521493
STEEL TEMP.2 = 44.1504517876
INPUT HEAT FLUX Q/A =
1530.25768255
HEAT LOSS THROUGH BOTTOM IN WATT
S = 3.79803340841E-2
CALCULATED TEMP. =
50.0973951777
HEAT LOSS 2 IN WATTS =
.233572101371
HEAT LOSS 3 IN WATTS =
.422660908653
TOTAL HEAT LOSS IN WATTS =
.694213424108
PERCENTAGE OF TOTAL HEAT LOSS =
9.49400470304
REAL HEAT FLUX = 1384.9749462 (W/m2)
HEAT TRA. COEF. = 724.768886138 (W/m2·C)
ELAPSED TIME(MIN.)=
2.9851333333
VERY GOOD
*****

```

Fig.3.9 Typical computer data output.

CHAPTER 4

ANALYSIS OF SWELLING

4.1 Analysis

As discussed in the literature survey (section 2.4), due to liquid boiling inside the tank, bubbles are formed on the surface and they raise the liquid level. In order to analyse the swelling phenomena inside the tank car, five assumptions have been made.

1) Bubble formation on the surface is evenly distributed over the surface.

2) Bubble rise velocity, V_b , given by Tong [17] is the same over the surface

$$V_b = 1.18 \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_l^2} \right]^{1/4} \quad (4.1)$$

Mass of the vapor generated by a given heat flux (kg/sec) is:

$$\dot{m}_v = \frac{\dot{Q}}{h_{fg}} \quad (4.2)$$

where $\dot{Q}$ and h_{fg} are total heat input (watts) and latent heat of vaporization (J/kg) respectively.

The volumetric rate of vapor generation (m^3/sec):

$$w = \frac{\dot{m}_v}{\rho_v} \quad (4.3)$$

where ρ_v is the density of the vapor.

The increase in volume due to vapor holdup is:

$$\Delta V' = wt \quad (4.4)$$

The time, t , taken for the vapor bubbles moving with a constant velocity, V_b , to rise to the top of the height is

$$t = \frac{H}{V_b} \quad (4.5)$$

As seen in Fig. 4.1, the height, H , is not constant but varies as a function of inclination angle, θ , due to tank geometry. To simplify the problem, another three assumptions are made:

3) Bubbles formed at the region A in Fig. 4.1, which is $0 \leq \theta < \gamma$, move vertically. In this region, the distance bubbles move can be expressed as:

$$H_1 = x + r \cos \theta \quad (0 \leq \theta < \gamma) \quad (4.6)$$

where x , r , and θ represent the liquid height above the centerline of the tank, radius of the tank, and the inclination angle which is an independent variable in the equation. This is illustrated in Fig. 4.2(a).

4) Bubbles formed at the region B, which is $\gamma \leq \theta < \pi/2$, move vertically and upon reaching the region C in Fig. 4.1, they slide along the surface. This is illustrated in

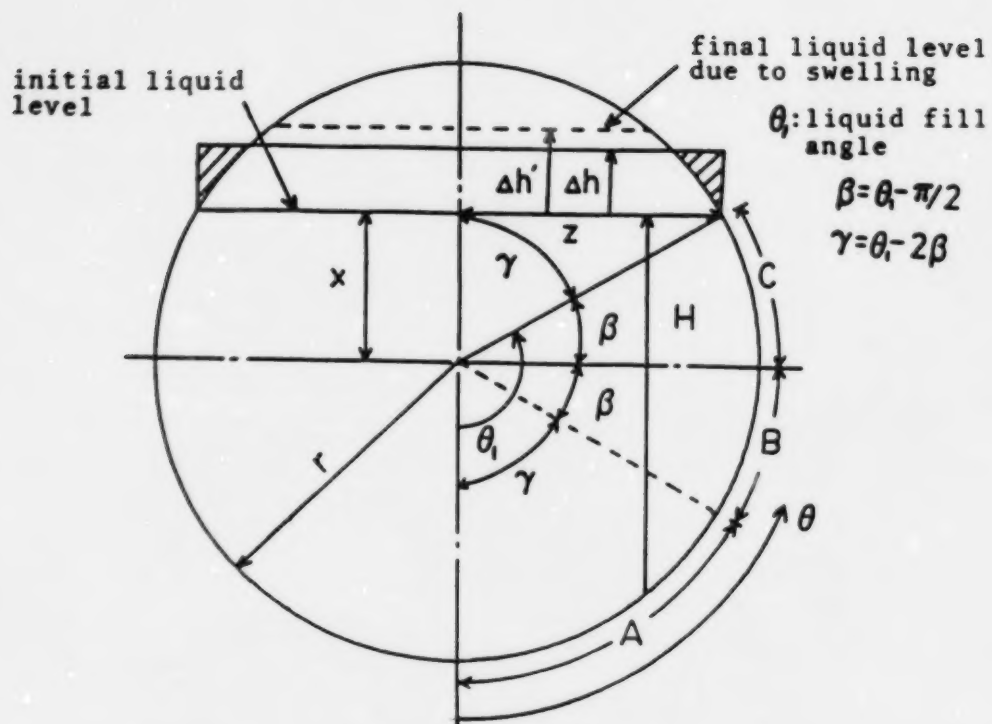


Fig.4.1 Idealized system for swelling.

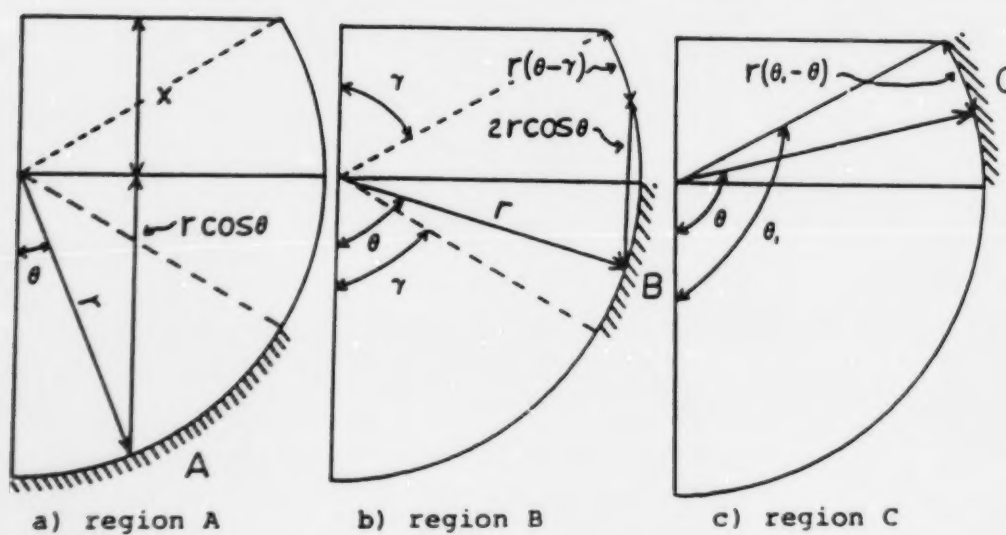


Fig.4.2 Heights at different regions.

Fig. 4.2(b). In this region, the distance bubbles move can be expressed as:

$$H_2 = 2 r \cos \theta + r(\theta - \gamma) \quad (4.7)$$

5) Bubbles generated at the region C, which is $\pi/2 \leq \theta < \theta_1$, slide along the curved surface. The distance bubbles pass becomes:

$$H_3 = r(\theta_1 - \theta) \quad (4.8)$$

Fig. 4.2 represents the eqns. (4.6), (4.7) and (4.8) as functions of θ .

At this stage, integration is done to evaluate an average height, H_{ave} , for the system. H_{ave} becomes

$$H_{ave} = \frac{\int_0^{\gamma} H_1 d\theta + \int_{\gamma}^{\pi/2} H_2 d\theta + \int_{\pi/2}^{\theta_1} H_3 d\theta}{\theta_1} \quad (4.9)$$

By substituting the values of H_{ave} and V_b , an average time, t_{ave} , can be calculated from eqn. (4.6). With this information an initial increased volume due to boiling, $\Delta V'$ can be evaluated and subsequently a swelled height, Δh can be determined.

As seen in Fig. 4.1, the actual swelled height is not Δh but $\Delta h'$ due to the cylindrical geometry of the tank. To account for the slashed area (Fig. 4.1), integration is

once again done to evaluate the actual swelled height, $\Delta h'$.

A typical example is given below for the UNB tank [20] in which R-11 serves as a working fluid and the radius of the tank is 0.19 m. At a pressure 2 bar inside the tank, the bubble rise velocity becomes 0.117 m/sec from the eqn. (4.1). The averaged distance, H_{ave} , is calculated from the eqn. (4.9) for the angle $\theta_1 = 13 \pi/18$ (130°) which represents the 85% liquid fill inside the tank. H_{ave} becomes 0.19 m. For the heat flux, $q/A = 10 \text{ kw/m}^2$, $\dot{m}_v = 2.6 \times 10^{-2} \text{ kg/sec}$, $w = 1.567 \times 10^{-3} \text{ m}^3/\text{sec}$ and $\Delta V' = 2.52 \times 10^{-3} \text{ m}^3$. With these values, the swelled height, Δh is calculated. $\Delta h = 1.76 \times 10^{-2} \text{ m}$. At this step, a modification is needed in eqn. (4.5) because bubbles move up the distance not H but $H + \Delta h$. Hence, a modified bubble holdup time, t_{mod} is 1.78 sec. With this t_{mod} value, the modified swelled height, Δh_{mod} , becomes $1.92 \times 10^{-2} \text{ m}$. As mentioned before, again integration is done to take into account the approximate nature of this estimate (Fig. 4.1). Finally the actual swelled height due to boiling, $\Delta h'$ becomes 2.0×10^{-2} .

4.2 Results and Discussion

The above theoretical swelling analysis has been applied to the UNB tank with different initial liquid fillings at $q/A = 16 \text{ kw/m}^2$ to compare it with the experimental data [20]. Table 4.1 shows the comparison

Table 4.1. Comparison between the theoretical and experimental swelling heights for UNB tank with different tank fillings
($q/A = 16 \text{ kw/m}^2$, at 2 bar)

Filling ξ	x (m)	θ_1 ($^\circ$)	H_{ave} (m)	t_{mod} (sec)	Δh (m)	$\Delta h'$ (m)	Δh_{exp} (m)
75	0.0772	114	0.1648	1.56	0.025	0.03	0.015
55	0.015	95	0.1298	1.2	0.015	0.017	0.013
25	-0.0772	66	0.0735	0.67	0.0066	0.0074	0.009

between the predicted and experimental results for the UNB tank. As seen in Table 4.1, the theoretically determined swelled height, $\Delta h'$ due to bubble holdup in the liquid column is a strong function of percentage of initial liquid filling. Swelling effect becomes vital especially for highly filled tanks because the free space for the disengagement of droplets is smaller.

Swelled height was also determined experimentally for the UNB tank at $q/A = 16 \text{ kw/m}^2$. These heights were illustrated as Δh_{exp} in Table 4.1. With lower liquid fillings (25%, 55%), the predicted values were quite compatible with the experimental data within the error of 18-24%. With 75% liquid filling, however, the predicted value was two times larger than the experimental data. This difference may be explained by the fact that while the theoretical model assumes saturation over the fluid inside the tank. However, in the actual case there is subcooling in the bulk liquid core and this subcooling becomes strong, especially for a highly filled tank. The subcooling causes the bubbles to condense on the surface so that the number of bubbles contributing to the swelling would be decreased.

Another swelled height determination was experimentally obtained for the T-vessel seen in Fig. 3.5. A transparent scale on the outside T-vessel was used to indicate the

variation in height due to bubble generation. Several photographs were taken with specific time intervals and timed video pictures were also recorded to determine the swelled height. Time averaged swelled height for the T-vessel at $q/A = 100 \text{ kw/m}^2$, 2 bar turned out to be 1 cm, which is compatible with the predicted value of 0.8 cm given by Singh and Gupta's model [16].

CHAPTER 5

RESULTS AND DISCUSSION

The pool boiling results obtained with the various surfaces as a function of surface orientation in Freon-11 and their discussion will be given in this chapter.

5.1 Experimental Results

Pool boiling heat transfer experiments with two plain surfaces (copper and mild steel) and four types of enhanced surfaces (UC, UNB #1, UNB #2 and a surface with Explosafe) were conducted in Freon-11. The surface orientation effect has been examined carefully. Figs. 5.1-5.8 represent these results and Table 5.1 illustrates the comparison of major parameters in the various surfaces. All tests were conducted at the pressure 2 bar except for the copper surface test in which data was obtained at three different pressures (1, 1.5 and 2 bar) to investigate the effect of pressure on boiling. Film boiling tests were carried out only for copper, UNB #1 and UNB #2 surfaces. The effects of orientation were studied for all surfaces except the UC surface. To investigate the effectiveness of Explosafe, two pieces of Explosafe (3/4" and 2" thick) were tested on the copper surface at 2 bar. Surface roughness tests and photographic studies of the surface cross sections were

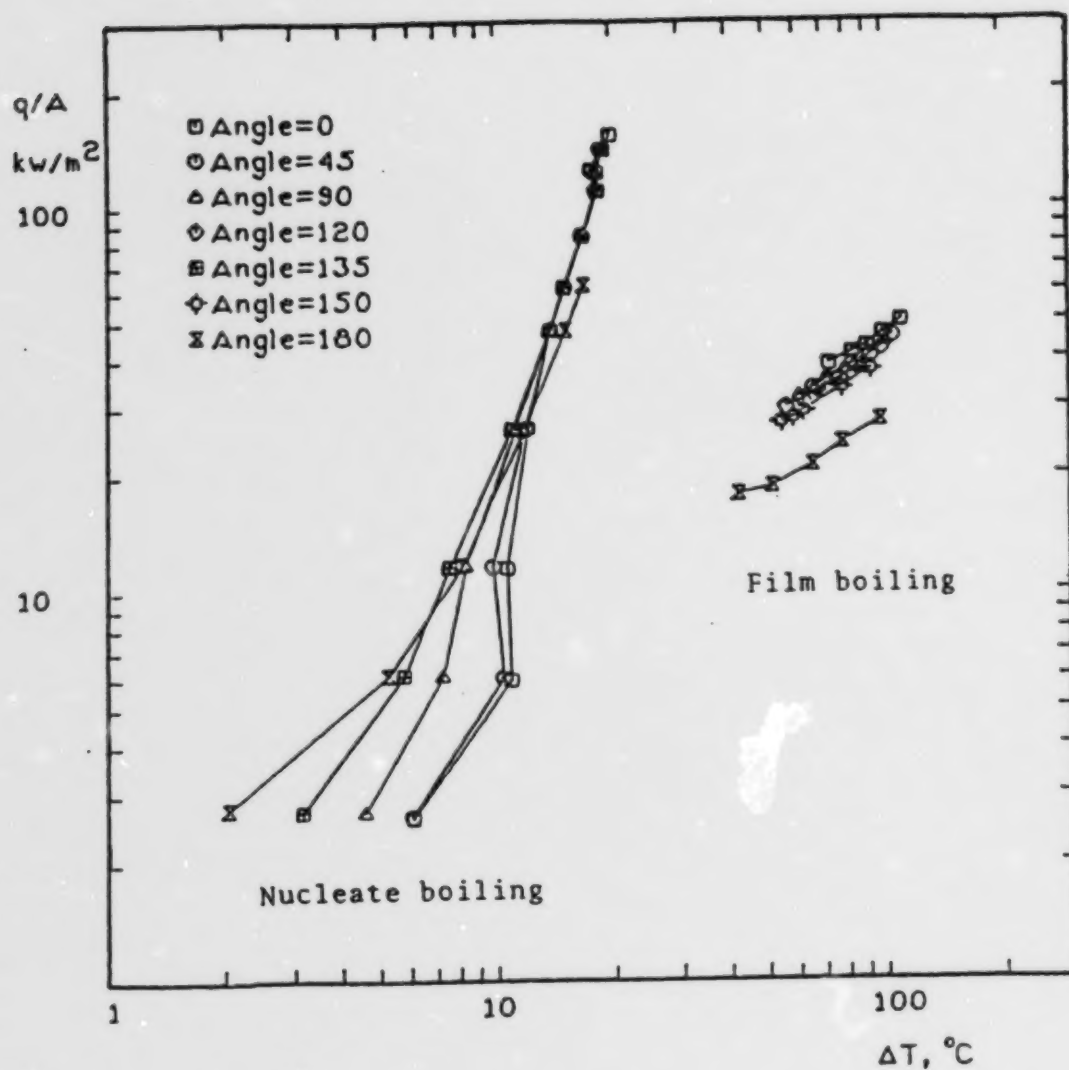


Fig.5.1 Pool boiling heat transfer with Freon-11 for copper surface at 2 bar.

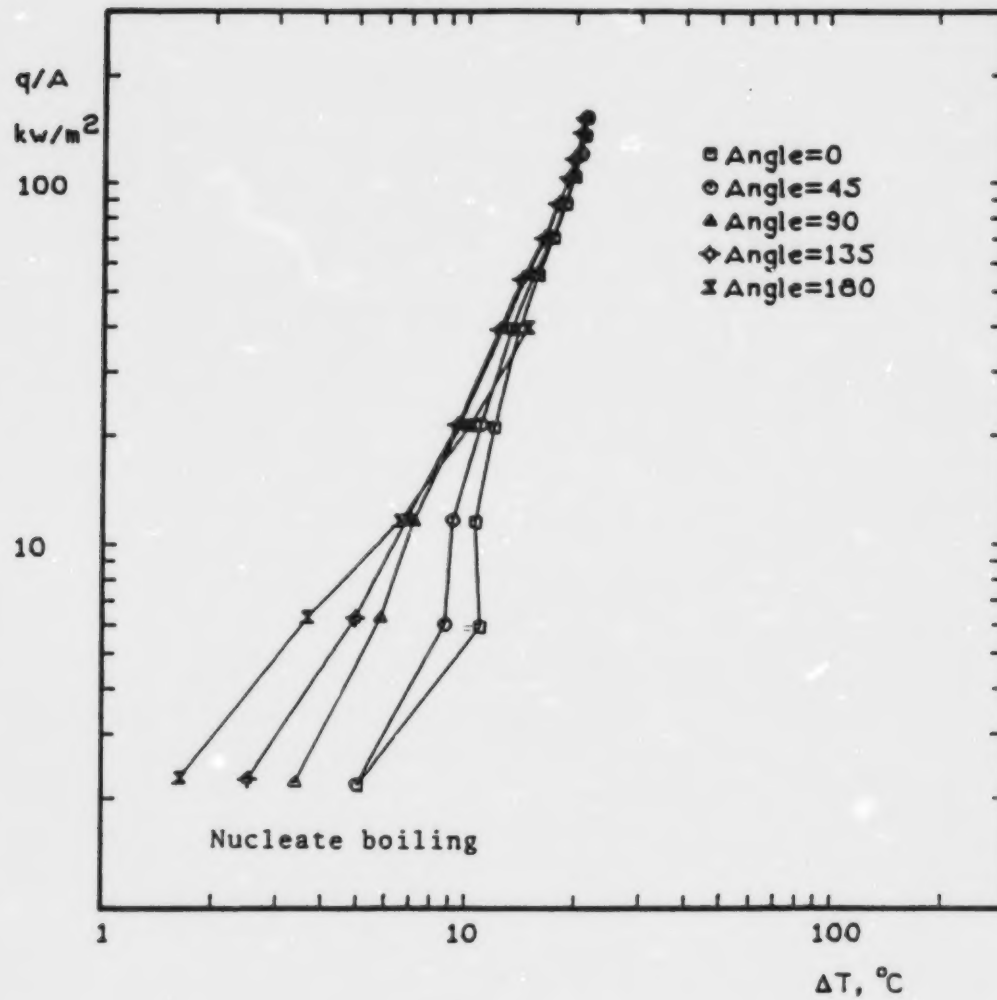


Fig.5.2 Pool boiling heat transfer with Freon-11 for mild steel surface at 2 bar.

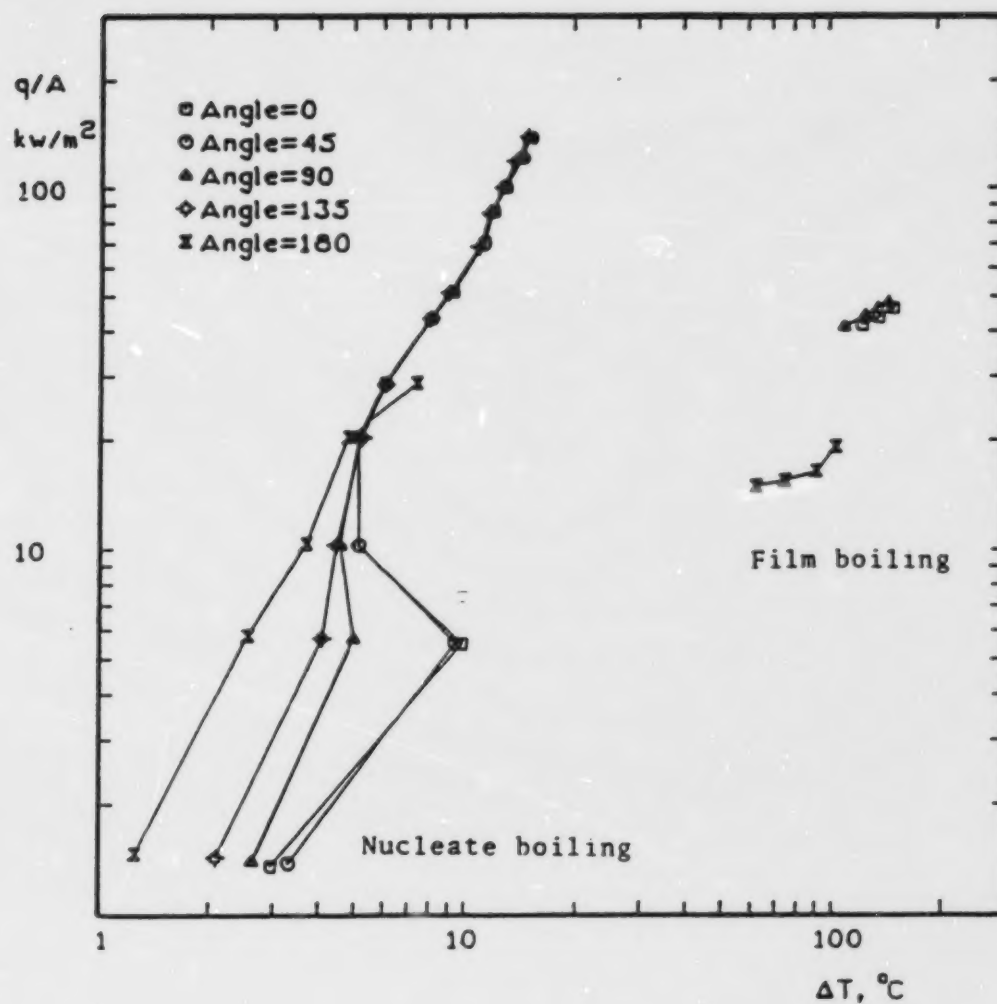


Fig.5.3 Pool boiling heat transfer with Freon-11 for UNB #1 surface at 2 bar.

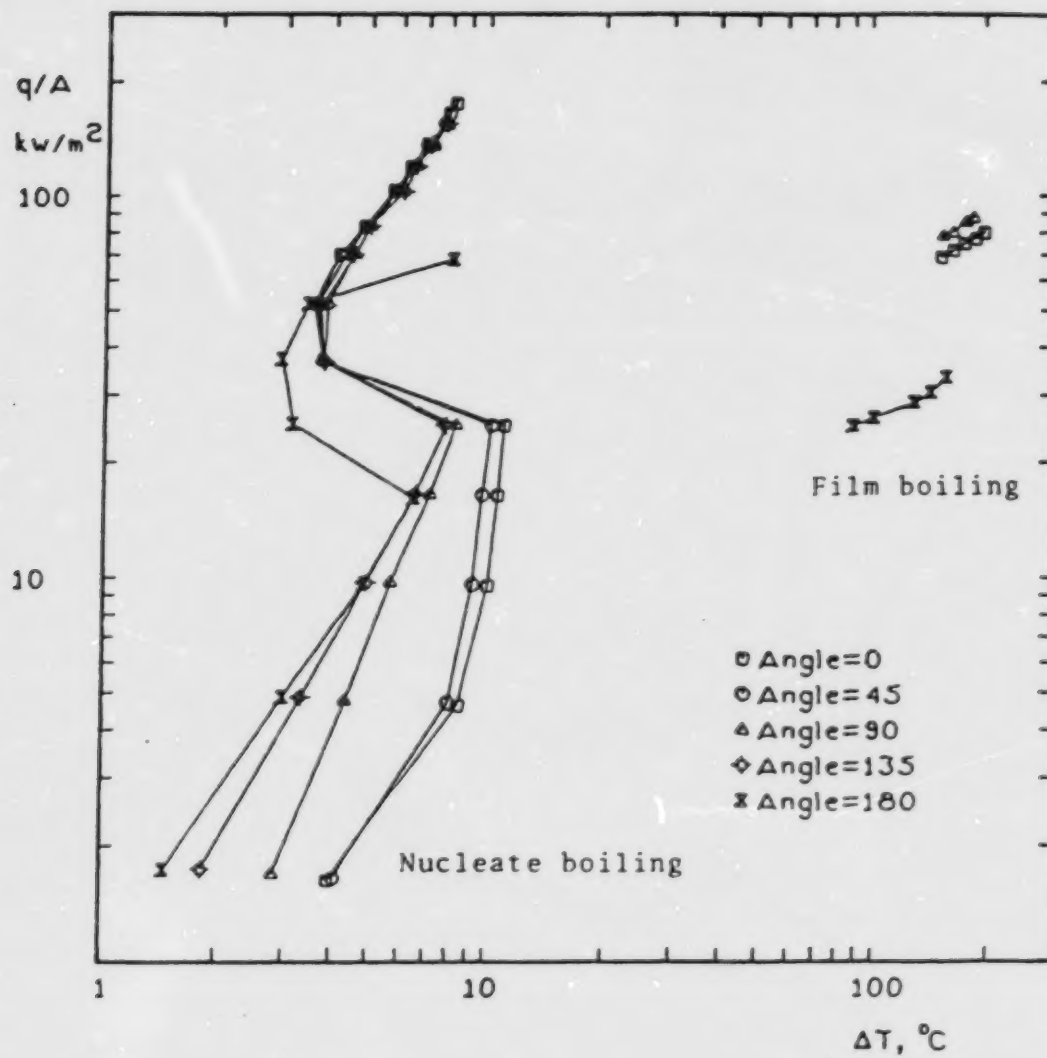


Fig.5.4 Pool boiling heat transfer with Freon-11 for UNB #2 surface at 2 bar.

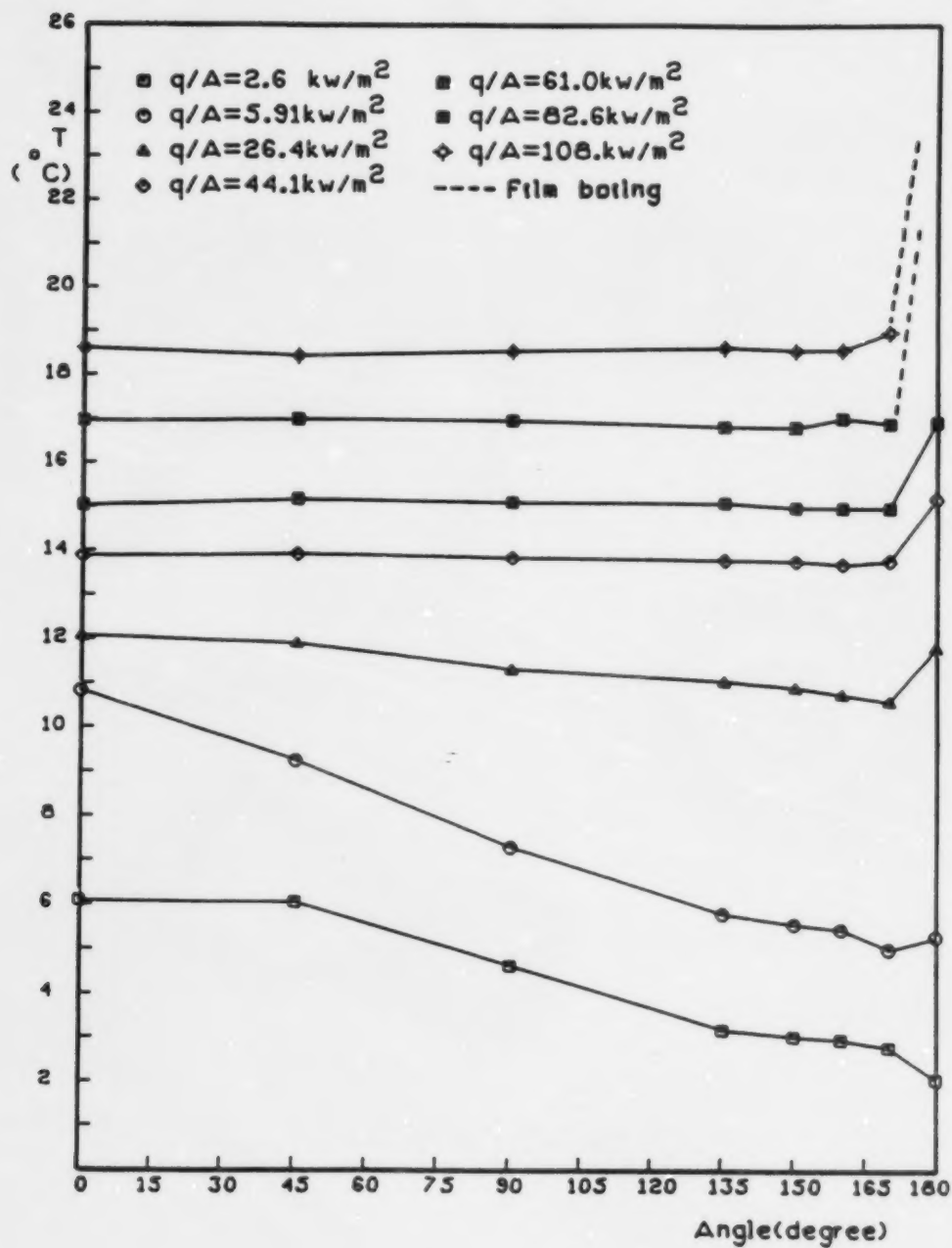


Fig.5.5 Effect of orientation for copper surface.

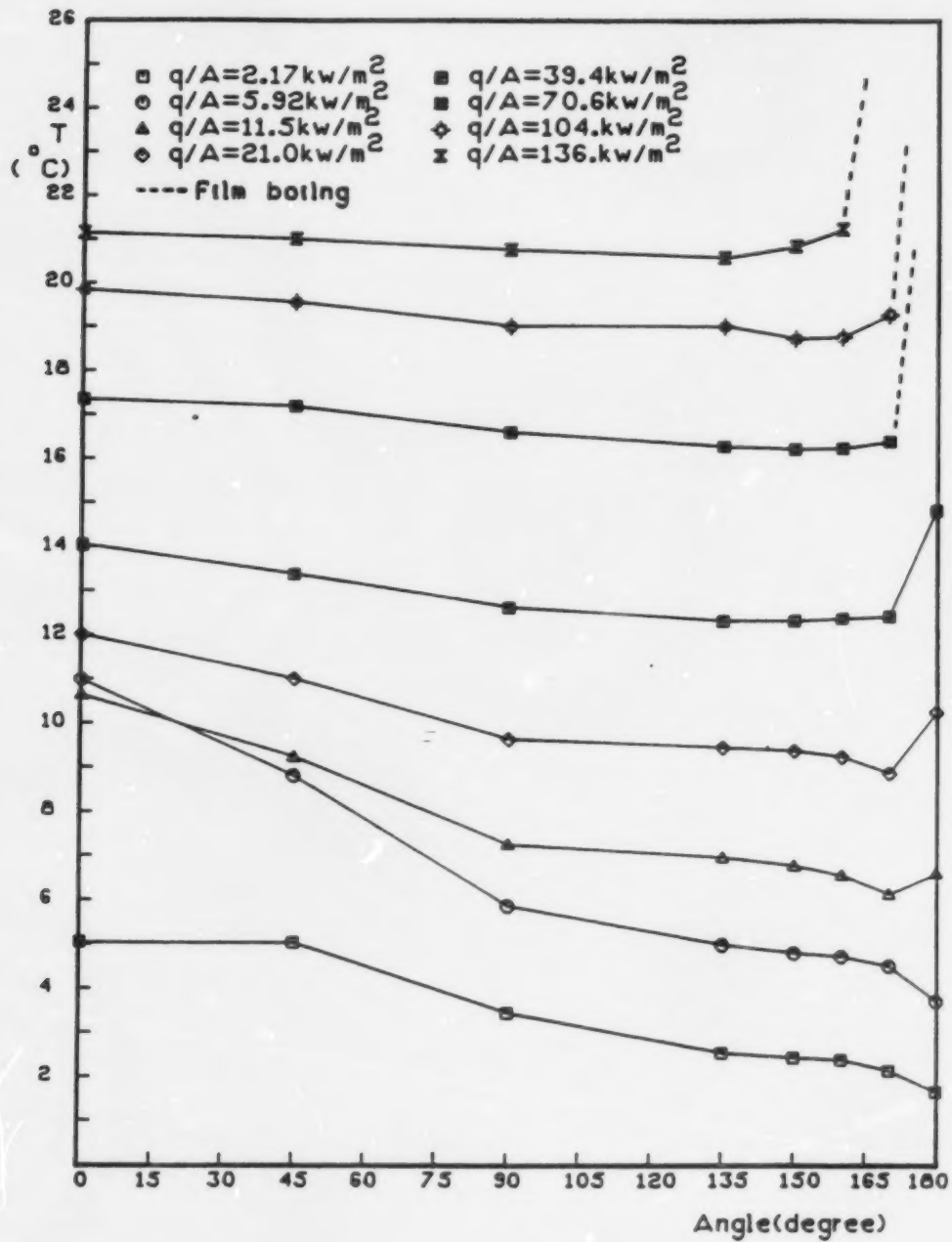


Fig.5.6 Effect of orientation for mild steel surface.

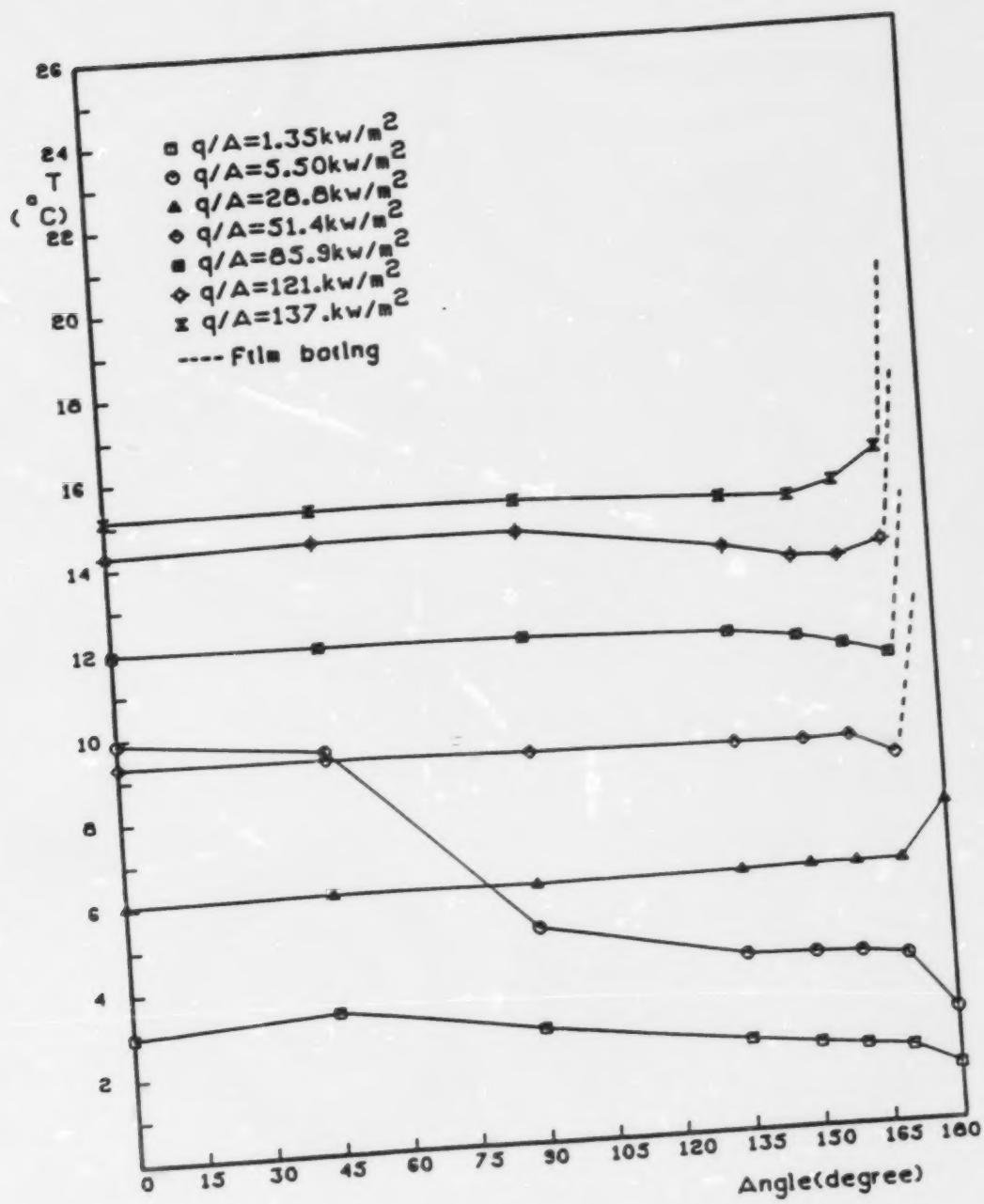


Fig.5.7 Effect of orientation for UNB #1 surface.

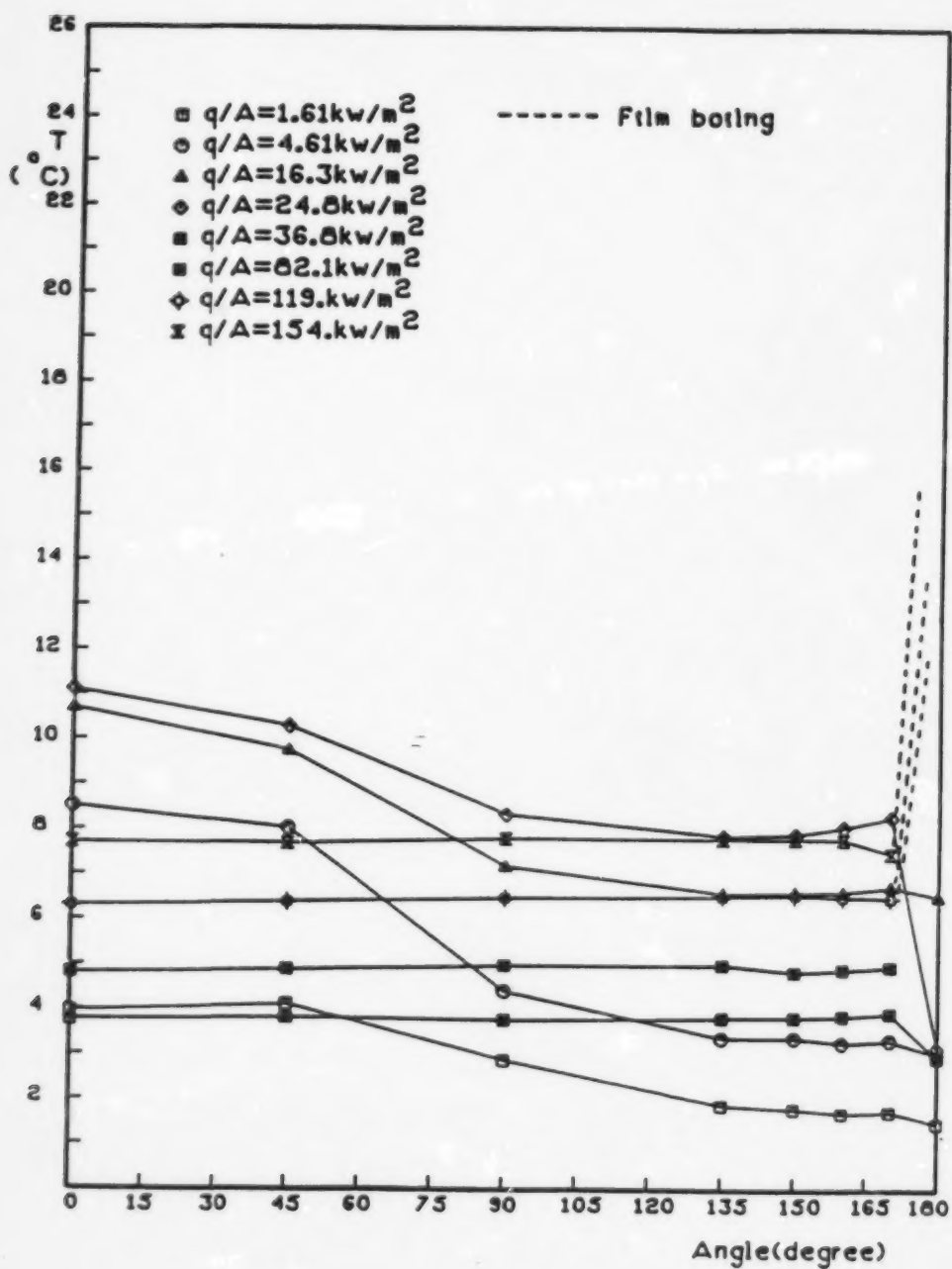


Fig.5.8 Effect of orientation for UNB #2 surface.

Table 5.1. Comparison of major parameters in various surfaces

parameters surfaces	NP _n	NP _f	(q/A) _ℓ (kw/m ²)	(ΔT) _ℓ (°C)	(q/A) _i (kw/m ²)	(ΔT) _i (°C)	(q/A) _h (kw/m ²)	(ΔT) _h (°C)	(q/A) _{min} (kw/m ²)	(ΔT) _{min} (°C)
Copper (2 bar)	11	5	2.6	6.07	5.91	10.83	152.3	19.98	38.4	72.8
Copper (1 bar)	13	0	2.67	7.53	6.1	13.0	150.3	24.5		
Copper with 3/4" Explosafe	12	0	2.47	5.49	5.81	10.3	139.3	17.9		
Copper with 2" Explosafe	12	0	2.57	5.44	6.31	10.62	148.5	18.6		
Mild Steel	12	0	2.17	5.05	5.92	10.98	152.2	21.33		
UC	12	0	1.4	2.94			158.7	15.9		
UNB #1	12	3	1.36	2.97	5.5	9.88	137.3	15.15	41.4	123.2
UNB #2	15	5	1.62	3.95	24.8	11.1	174.6	8.18	68.8	151.9

(NP represents the number of data points and the subscripts n, f stand for nucleate and film boiling respectively and ℓ, i and h mean the lowest, incipient and highest boiling points in nucleate boiling and 'min' represents the minimum boiling point in film boiling.)

done in an attempt to explain the bubble nucleation mechanism and film boiling results. Finally, all results in nucleate boiling regime were correlated by using the Rohsenow's pool boiling equation.

5.2 Discussion

5.2.1 Single Phase Natural Convection

Natural convection has been observed for all surfaces except UC as shown in Fig. 5.9. The heat flux range associated with natural convection was approximately $1.5 \sim 10 \text{ kw/m}^2$ as illustrated in Table 5.1.

There is a distinct difference in natural convection curves between UC and other surfaces as shown in Fig. 5.9. The heat transfer coefficient for UC is 2.6 times higher than those of other surfaces (copper, mild steel, UNB #1 and UNB #2).

This phenomena may be explained as follows: In UC, even at low heat fluxes, small nucleation sites are activated and hence produce small bubbles to take the heat away. The other surfaces do not have such small artificial pores so that it takes longer for those surfaces to activate nucleation sites. This supports the agitation theory [21, 22] that the observed increase in heat flux density for a given surface is a result of increased agitation associated with the liquid behind the wake of departing bubbles.

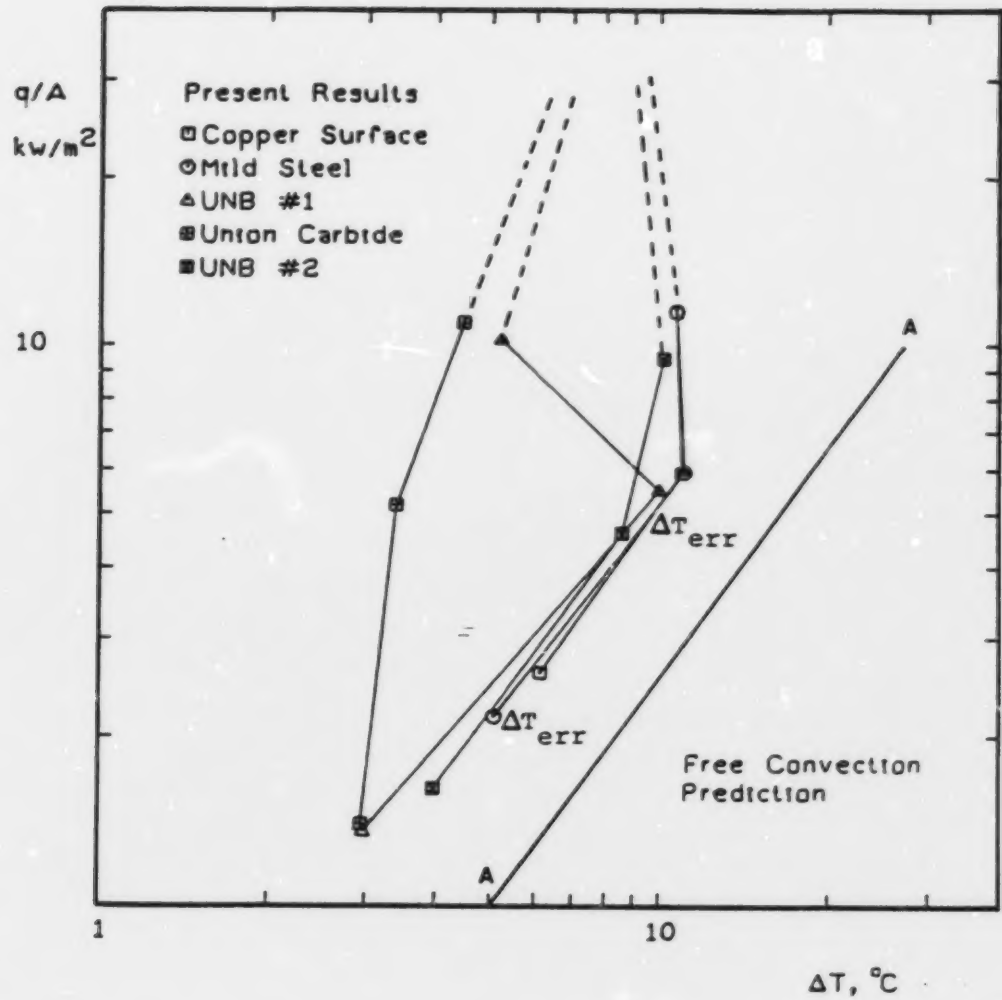


Fig.5.9 Comparison of the present results with theoretical prediction in free convection at horizontal case.

The present results in natural convection for plain horizontal upward facing surfaces were compared to an empirical correlation by King [23]. Curve A in Fig. 5.9 represents this free convection prediction. There is $3^\circ \sim 5^\circ$ temperature difference, ΔT_{err} in Fig. 5.9 between the correlation and the experimental data for plain surfaces. The slopes, however, of both empirical and experimental curves are the same. There are three major factors responsible for this discrepancy. a) the sub-cooling due to the heat loss through the glass T-vessel to the surroundings caused a $0.6 \sim 0.7^\circ\text{C}$ of temperature difference; b) further experiments at one bar verified that the auxiliary heaters induced convection which account for a $2^\circ \sim 3^\circ\text{C}$ of temperature difference. This will be discussed later in detail when we consider the effect of inclination; c) uncertainties in the correlation [23] are believed to compensate for the remaining part.

The present results (Figs. 5.5 ~ 5.8) exhibit that in the natural convection regime, as the inclination angle increases, so the superheat continuously decreases. This trend is in good agreement for the range of 0° (horizontal) to 90° (vertical) but differs from that of previous investigations [7, 8] for the range of 90° to 180° (facing downward), where the superheat increases according to the angle resulting in the decrease in heat transfer

coefficient.

Further experiments at 1 bar with UNB #2 surface were, therefore conducted to examine this phenomena. With the same heat flux $q/A = 1 \text{ kw/m}^2$, natural convection curves were obtained as a function of inclination angle with and without operating the auxiliary heaters. The power input to the auxiliary heater was 40 watts. Fig. 5.10 illustrates the results as a function of angle. It is noted that the superheat difference between the curve A and B in Fig. 5.10, ΔT_{conv} is a strong function of the inclination angle showing a continuous increase according to the angle. This may be simply explained in such a manner that as the angle increases, bubbles from the auxiliary heaters can have better access to the surface disturbing the natural convection boundaries. With this result, it is clear that the induced convection effect by the auxiliary heaters has a strong influence in the natural convection regime.

The present results for UNB #2 surface at 1 bar without turning on the auxiliary heaters were compared with the empirical correlation by Fujii and Imura [8]. Once more data at the heat flux, $q/A = 1.75 \text{ kw/m}^2$ was obtained for this purpose. Fig. 5.11 shows the comparison between the empirical and experimental results. In determining the curve for the empirical correlation, eqns. (2.1) ~ (2.5) were used for each domain and all

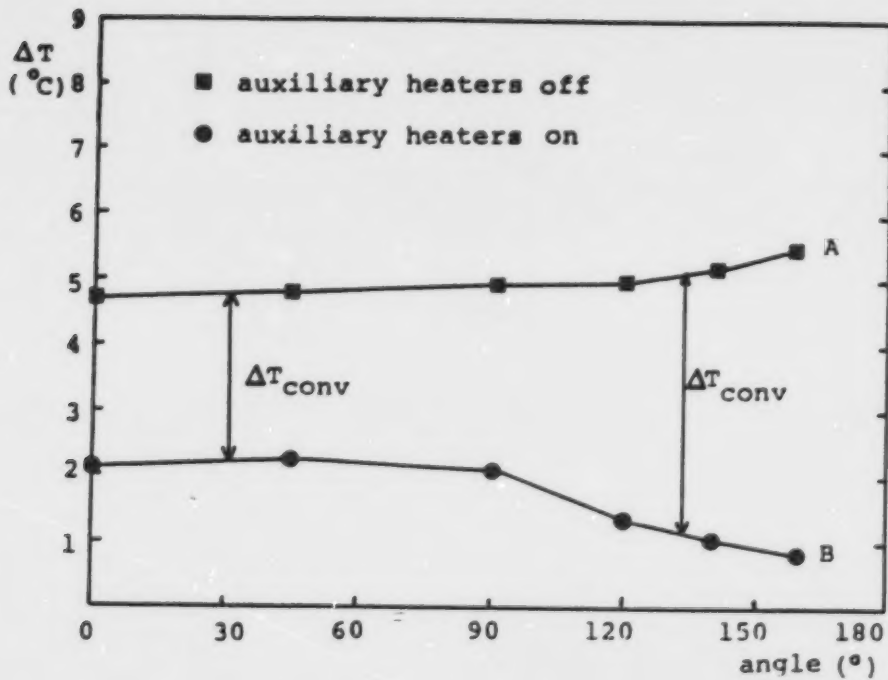


Fig.5.10 Convection effect by auxiliary heaters as a function of the inclination angle (UNB #2 surface, 1 bar, 1kw/m^2 , power input to the auxiliary heaters is 40 watts).

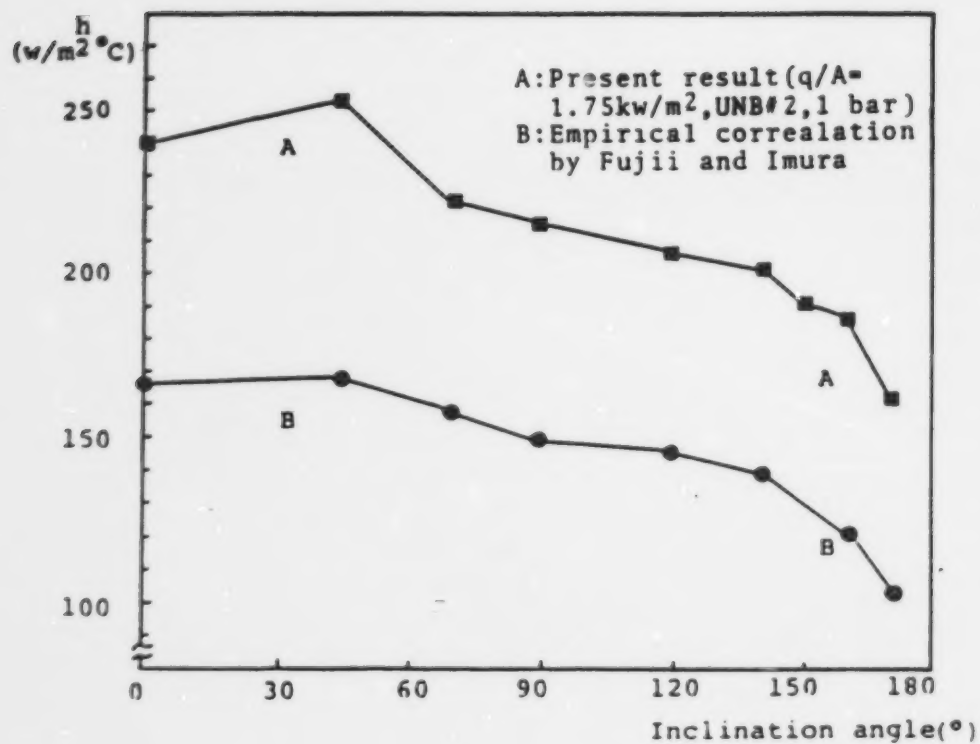


Fig.5.11 Experimental data with UNB #2 surface in natural convection region and empirical correlation of Fujii and Imura[8].

properties of liquid Freon-11 were taken from [24]. While the present study utilizes a circular section, Fujii and Imura used a square as their test section. To account for the geometry effect in evaluating the Grashof number, which includes a characteristic length of the surface, $0.9 d$ (d is the diameter of the test section) was used as the characteristic length as suggested by Holman [25]. Fig. 5.11 illustrates the result, where the heat transfer coefficient is represented as a function of the inclination angle, θ . In essence, the shapes of both curves are almost the same but the curve A (present result) is shifted to the top indicating 40 ~ 45% better heat transfer. This may be explained by the fact that Fujii and Imura's experimental apparatus restricted the main flow two dimensionally but the present apparatus permitted three dimensional flow such that the boundary layer gets thinner and hence heat transfer is improved. Out of this investigation, it is concluded that as the inclination angle increases, heat transfer is retarded in the natural convection regime, which in essence, coincides with the previous investigations [7, 8, 9].

5.2.2 Hysteresis Effects in Incipient Boiling

All surfaces except UC showed the existence of incipient boiling points and marked the 'knee' on the boiling curve as shown in Figs. 5.12 ~ 5.14.

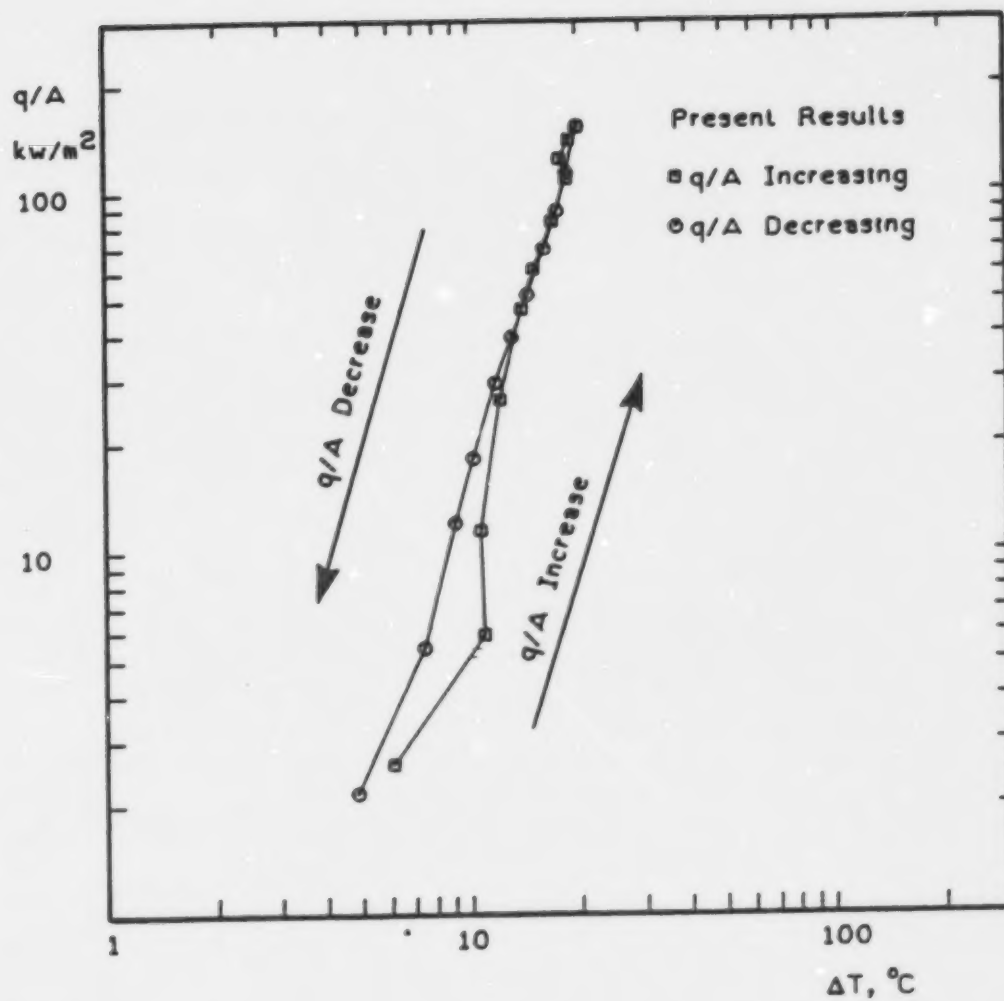


Fig.5.12 Hysteresis effect for copper surface at 2 bar in R-11.

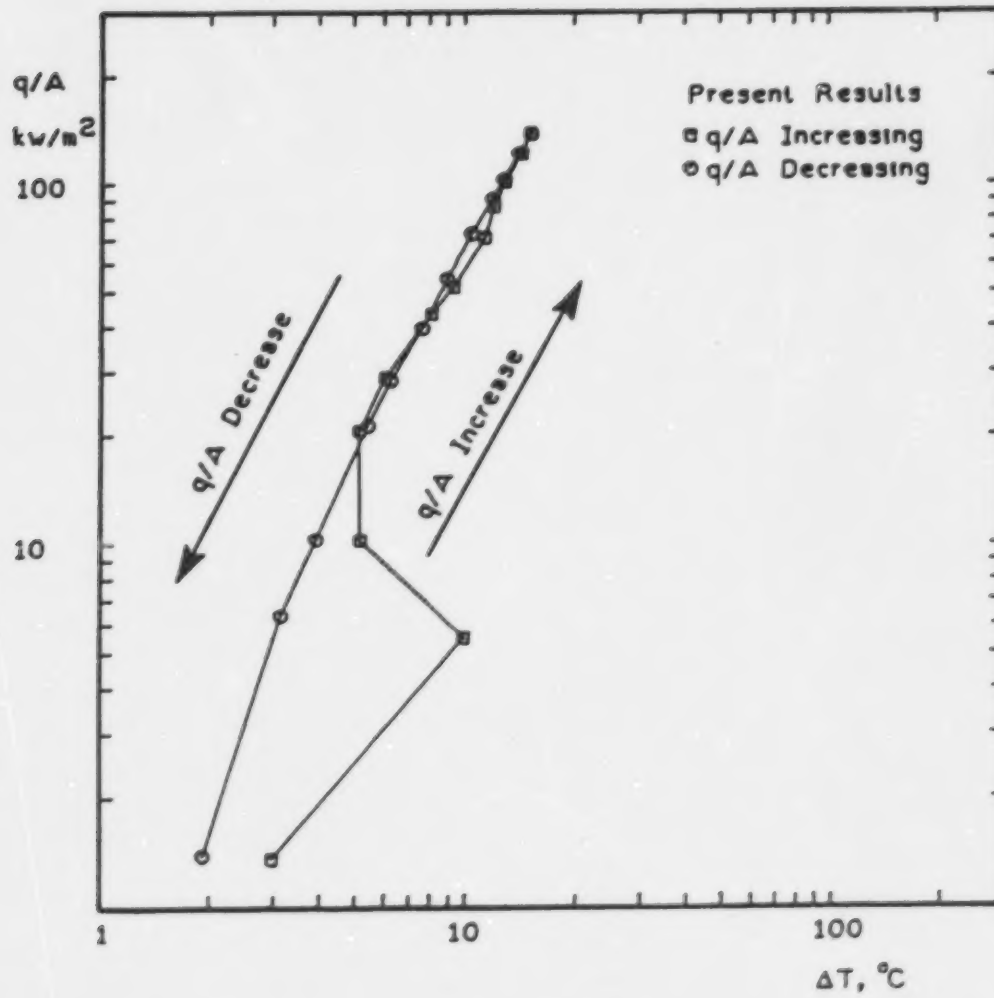


Fig.5.13 Hysteresis effect for UNB #1 surface at 2 bar in R-11.

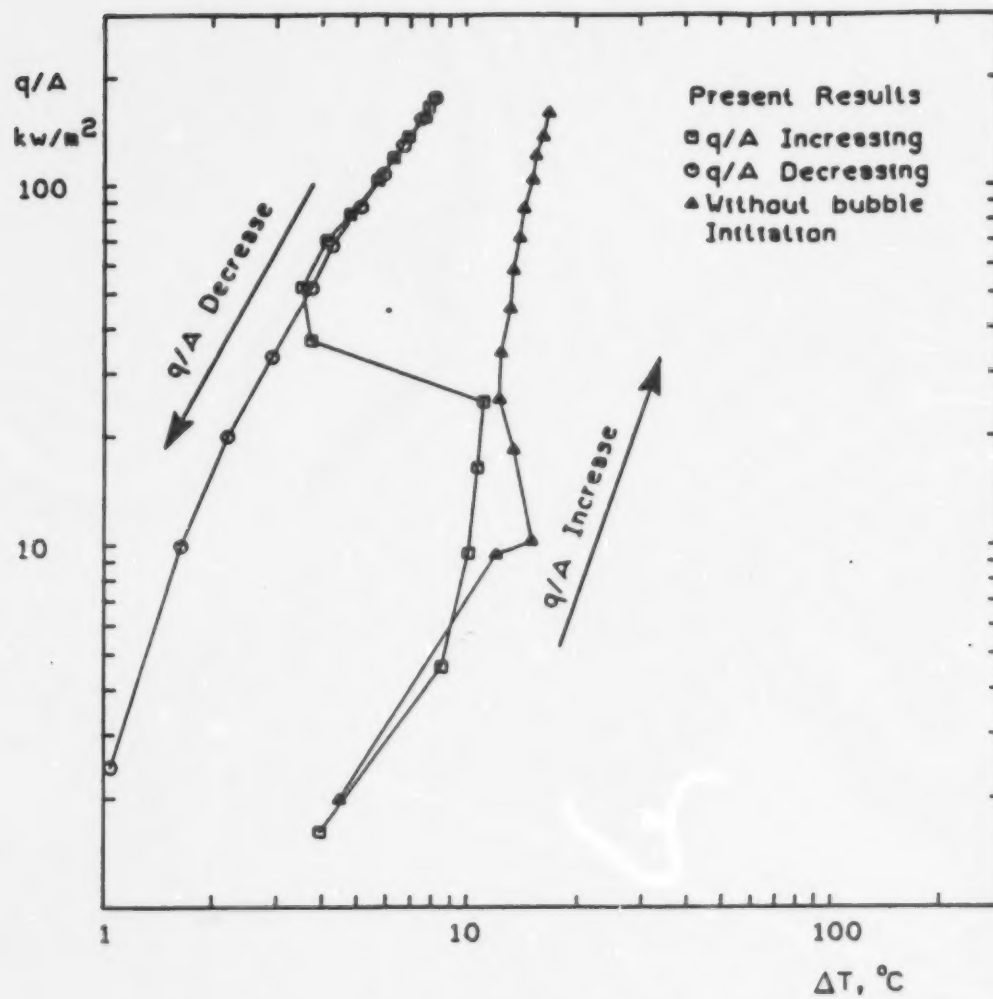


Fig.5.14 Hysteresis effect for UNB #2 surface at 2 bar in R-11.

When the level of the superheat required for incipient boiling is reached, any small increase in the heat flux will result in a vigorous boiling accompanied by a drop in the surface temperature. The value for incipient boiling superheat obtained in the present study with a copper surface at 1 bar is 13°C as illustrated in Table 5.1. It was dependent upon the system pressure in such a manner that this value decreases to $10\text{--}11^{\circ}\text{C}$ at 2 bar. Note that the value for the incipient boiling point is independent of the surface material as long as the surface finishes are the same as shown in Fig. 5.9 (copper and mild steel curves).

An analytical model to predict the superheat corresponding to the incipient boiling point was reported by Chen [26]. The minimum superheat predicted by Chen for R-11 is 14.5°C for a deactivation condition at 21°C , 1.3 bar. This is slightly larger than the measured experimental value here of 13°C . Abdelmesshi et al.'s [27] experimental value for flow boiling in Freon-11 is $14\text{--}16^{\circ}\text{C}$. Out of these results it is thought that the incipient boiling point in Freon-11 at 1 bar is approximately $13\text{--}16^{\circ}\text{C}$.

Figs. 5.12 ~ 5.14 also compare the results for two different cases. Heat flux was increased from $(q/A)_{\ell}$, the lowest data point, to $(q/A)_{\text{h}}$, the highest point and then continuously decreased to $(q/A)_{\ell}$. These are represented as q/A increasing and q/A decreasing respectively

in Figs. 5.12 ~ 5.14. In the q/A increasing case no bubbles were formed on the surface below the incipient boiling point. However, in the q/A decreasing case two distinct effects were observed. Firstly, there was no shift in the incipient boiling position and secondly the superheat required to sustain boiling decreased with progressive decrease in the heat flux. Therefore, the effects of past history known as 'hysteresis' is very well illustrated. This hysteresis effect comes from the activation of cavities as the heat flux decreases which had been inactive while the heat flux increased from low heat flux. Figs. 5.13 and 5.14 show typical figures for hysteresis effect with UNB #1 and UNB #2. These two surfaces are characterized by large scale boiling curve hysteresis, an effect which has also been reported by Bergles and Chyu [15].

5.2.3 Nucleate Boiling

After passing the incipient boiling points, all surfaces undergo fully developed boiling. The heat flux range related to nucleate boiling is approximately $10 \sim 170 \text{ kw/m}^2$ as shown in Table 5.1. As seen in Fig. 5.15, the copper surface showed better heat transfer performance than mild steel. For instance, at $q/A = 152 \text{ kw/m}^2$, the heat transfer coefficient for copper ($7607 \text{ w/m}^2\text{°C}$) is 7% higher than that for mild steel ($7126 \text{ w/m}^2\text{°C}$), which implies that the thermal conductivity of the heating

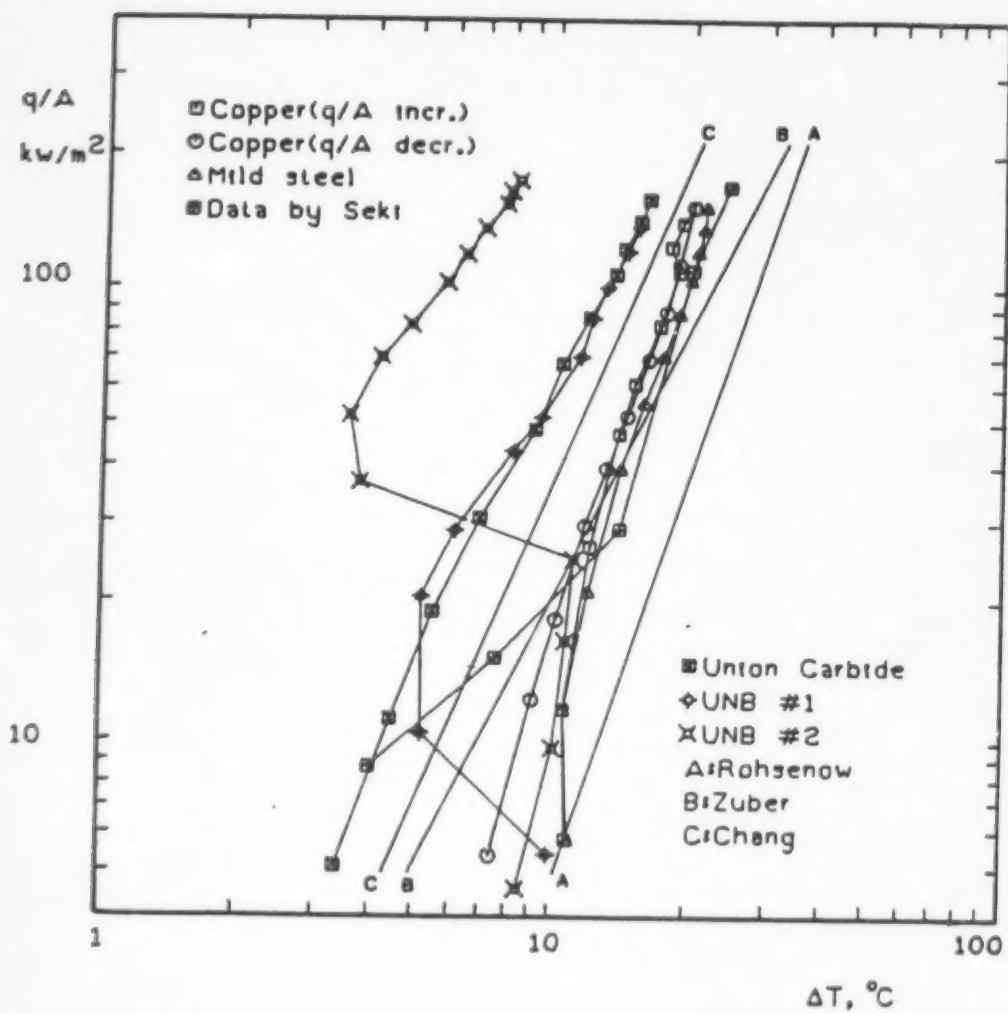


Fig.5.15 Comparison of the present results with available data and correlations in nucleate boiling region.

surface is one of the variables in nucleate boiling.

For enhanced surfaces, the boiling curves are shifted to the left, which indicates that these surfaces have higher heat transfer coefficients. Table 5.2 compares the heat flux and heat transfer coefficients with given superheats for the different surfaces tested. Enhanced surfaces show much better heat transfer than plain surfaces as illustrated in Table 5.2. For instance, with the given superheat, 8°C , UC and UNB #1 have ten times greater heat transfer coefficients as compared to plain surfaces. Surprisingly, however, UNB #2 far surpasses their values showing a 42.5 times higher value than that of ordinary surfaces.

This heat transfer improvement may be explained on the basis of the surface roughness tests performed using a Talysurf 5. Table 5.3 illustrates the surface roughness test results and Fig. 5.16 shows the surface profiles for the various specimens.

UNB #2 records the highest roughness, Ra and high spot count, HSC. On the other hand, UC and UNB #1 have smaller values of Ra as compared to that of UNB #2 but still these two surface Ra values are 5 ~ 6 times greater than that of plain surfaces. Boiling curves and this roughness test result coincide with previous investigations by Corty et al. [28] and Kopp et al. [29] saying that

Table 5.2. Comparison in heat flux and heat transfer coefficient for given superheats with different surfaces

$\Delta T(^{\circ}\text{C})$	Copper	Mild Steel	UC	UNB #1	UNB #2
$q/A \text{ (kw/m}^2\text{)}$					
5	2.1	2.1	14	15	84
8	4	4	40	42	170
16	68	56	160	160	
$h \text{ (w/m}^2\text{ }^{\circ}\text{C)}$					
5	420	420	2800	3000	16800
8	500	500	5000	5250	21250
16	4250	3500	10000	10000	

Table 5.3. Surface roughness parameters obtained with a Talysurf 5 from different surfaces

Parameters	Copper	Mild Steel	UC	UNB #1	UNB #2
S_m (μm)	45.1	56.66	357.4	314.4	211
Δq	0.39	0.377	0.7	0.7	0.67
Rsk	0.366	-0.146	0.126	-0.09	0.05
HSC	0	0	2.86	4	4.71
R_p (μm)	6.2	6.25	30.4	20.1	31.5
R_{max} (μm)	8.8	9.76	49	41	60.5
R_a (μm)	1.56	1.51	9.5	6.61	11.7
Rat	1.076	1.071	1.245	1.245	1.224

- R_a : R_a is the universally recognized and most used, international parameter of roughness. It is the arithmetic mean of the departures of the profile from the mean line.
- R_{max} : R_{max} is the maximum peak-to-valley height of the profile within the sampling length L .
- R_p : R_p is the maximum height of the profile above the mean line within the assessment length.
- Rsk : Rsk is a skewness which is a measure of the asymmetry of the amplitude distribution curve above the mean line.
- HSC : The high spot count is the number of complete profile peaks projecting above the mean line.
- S_m : S_m is the mean spacing between profile peaks at the mean line, measured over the assessment length.
- Δq : This is the rms slope of the profile throughout the assessment length.
- Rat : Rat is the ratio of the actual profile length to nominal profile length. $\text{Rat} = 1 + 1/2(\Delta q)^2$.



(a) Mild steel



(b) Union Carbide



(c) UNB surface #1



(d) UNB surface #2

Fig.5.16 Surface profiles obtained with a Talysurf 5 for different specimens (horizontal = $200\mu\text{m}$, vertical = $100\mu\text{m}$, and detailed surface parameters are given in Table 5.3).

a rough surface has superior heat transfer to a fine one.

The role of the surface roughness on boiling may be explained in the following fashion. In terms of the average slope of the surface profile, the three rough surfaces have relatively higher values compared to plain surfaces. The average slope inherently represents the ratio of the actual profile length to nominal profile length. The steeper the slope of the profile, the longer the actual length with its nominal length. Increase in actual length means that a surface has a larger actual heat transfer area which potentially provides higher site density or number of active sites per unit area. As Yamagata [30] has shown by the correlation

$$q/A \propto \Delta T_n^{a,b} \quad (5.1)$$

where $\Delta T = T_w - T_{sat}$ and n is the site density, q/A is directly proportional to the site density of a surface. This is believed to be the strongest reason for the high heat transfer coefficient in rough surfaces.

UNB #2, despite its great heat transfer characteristics, however, has its own drawback in that it is very difficult to activate the nucleation sites resulting in the largest knee at high superheat in the boiling curve as shown in Fig. 5.14. On the other hand, UC and UNB #1 have their own capabilities to activate the cavities easily at low heat

flux.

These differences among UNB #1, #2 and UC in terms of activating nucleation sites can be explained on the basis of photographic studies of the surface cross sections under the microscope. Fig. 5.17 illustrates the one hundred times magnified cross sectional views of the various surfaces. As seen in Fig. 5.17(a), the plain surface doesn't have artificial pores except its own pits and scratches. On the other hand, the UC surface has the largest number of small pores. UNB #1 has a smaller number of large pores compared to the UC surface. But UNB #2 doesn't have many pores but has a number of long inter-connecting channels. At the initial stage, liquid penetrates into the large pores and interconnecting channels causing the possible nucleation sites to be flooded out. For the very small pores, this is not the case because liquid cannot wet the pores so that vapor might be entrapped to generate bubbles.

As for the UC surface, it has small nonwetable pores which cause fully developed boiling even at a very low heat flux and hence avoids the 'knee' in the boiling curve. UNB #1, however, has larger pores which were initially flooded by the liquid but on reaching the condition, that is, the temperature of the pores becoming hot enough to evaporate the liquid around, the superheat suddenly dropped



(b) Union Carbide



(d) UNB#2



(a) Mild steel



(c) UNB#1

Fig. 5.17 Microscopic views of cross sections for various specimens (100X).

and the results are quite compatible with that of UC, a fact which implies that the porosities of the two surfaces are almost the same but only the pore sizes are different.

As for UNB #2, it takes longer for the metal coating to be activated because of a shortage in nonwetted pores as well as liquid flooding in the channels. As mentioned before, however, once the interconnecting channels are superheated, it has the highest performance. This phenomena may be explained by the fact that it has the largest actual heat transfer area due to the multi-layered rough coating and hence the site density increases very much as discussed before.

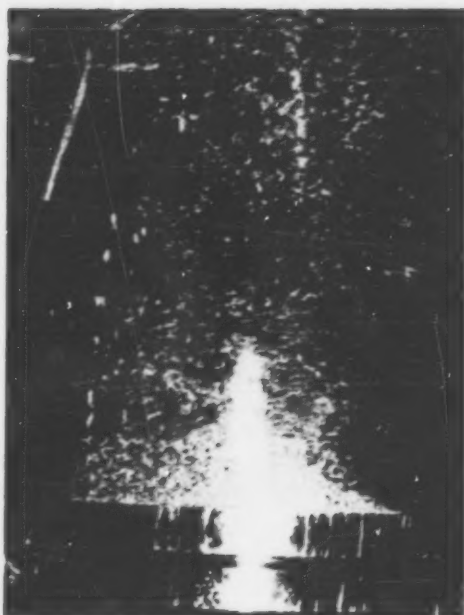
Boiling curves for the horizontal case with various surfaces were compared with the theoretical prediction as shown in Fig. 5.15. None of the three well known equations correlates the data well. Rohsenow's equation [32] (curve C) overpredicts the actual phenomena. Zuber's equation [33] (curve B) overpredicts below $q/A = 40 \text{ kw/m}^2$ and goes on to underpredicting beyond 40 kw/m^2 . This fact supports the premise [3] that any boiling correlation which embodies only liquid and vapor properties cannot be a universal one. Besides, those equations in Fig. 5.15 are no longer valid to correlate the data from the enhanced surfaces because those equations were derived on the basis of plain surfaces.

Out of these results it is very clear that some modifications are needed in those equations to correlate the data for both plain and enhanced surfaces.

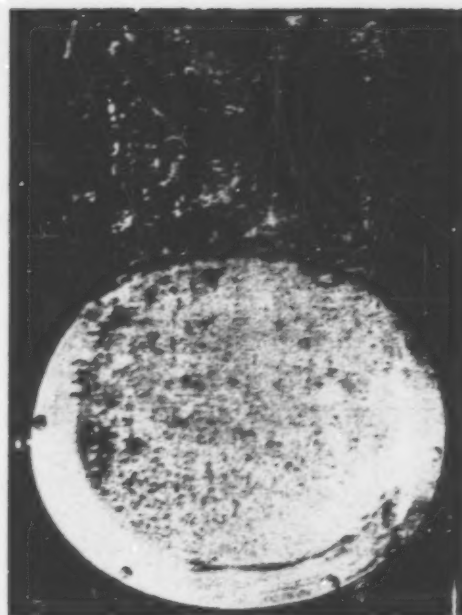
Chang's and Zuber's equations have fixed constants and exponents so that there is no room for modification. However, in Rohsenow's equation this is not the case. It has the unique flexibility that the constant ' C_{sf} ' and exponent ' r ' can be changed according to the surface characteristics and surface-liquid combination. In this sense, Rohsenow's equation is a universal one and is used here to correlate the results (section 5.2.7).

The effect of orientation in nucleate boiling was carefully studied. Figs. 5.5 ~ 5.8 illustrate the results for the various surfaces and Fig. 5.18 shows typical nucleate boiling at different inclination angles for the copper surface. In the nucleate boiling regime with plain surfaces, the heat transfer coefficient increases 30 ~ 40% as the angle changes from 0° to 165° in the heat flux range 10 ~ 40 kw/m^2 . Beyond this range, however, the superheat remains almost constant regardless of surface orientation except for the facing downward case. The results at high heat flux deviate from the previous investigations in which the result for the vertical case shows the smallest superheat. This discrepancy may be caused by the edge effects discussed in Chapter 3.

For the facing downward case the superheat required



(a) angle=0 (horizontal)



(b) angle=90 (vertical)



(c) angle=135



(d) angle=180 (facing down)

Fig.5.18 Typical nucleate boiling at different surface orientations.

rises substantially indicating that the heat transfer coefficient is decreased. This can be explained by the fact that when the heating surface is facing downward, bubbles are apt to collect on the underside due to buoyancy forces. It thus appears that the gross convection currents are not sufficient enough to sweep those bubbles away. As a consequence, the bubble motion does not add to the convection mechanism as it does when the surface is in other orientations.

The enhanced surfaces, UNB #1 and UNB #2 reveal almost the same trend at high heat flux that the superheat keeps constant and increases for facing downward as shown in Figs. 5.7 and 5.8.

The effect of orientation, however, plays an important role in activating nucleation sites at the low heat flux range. Regarding UNB #1 (Fig. 5.7), the superheat required for $q/A = 5.5 \text{ kw/m}^2$ is 10°C at the horizontal case and it decreases very rapidly according to the inclination angle and becomes 3°C (facing downward). The heat flux was increased to 28.8 kw/m^2 but the superheat required for the horizontal case is 6°C which is 40% lower than that for $q/A = 5.5 \text{ kw/m}^2$ and it keeps constant regardless of the angle and increases for facing downward. This is well illustrated in Fig. 5.7. Fig. 5.8 also illustrates the same trend for UNB #2.

Further experiments were carried out in an attempt to explain this phenomena by comparing with the boiling curve

on the horizontal surface for UNB #2. Fig. 5.14 illustrates that the boiling curve obtained for the horizontal surface with continuously increasing heat flux (without bubble initiation) is very different from the one in which the inclination angle was changed from horizontal to facing downward as the heat flux increased (q/A increasing). In the former (without bubble initiation) the boiling curve does not show a sudden superheat drop all along the heat flux ranges except at the incipient boiling point, which is basically the same trend as that for plain surfaces. In the latter, however, the superheat suddenly drops from 10°C to 3.5°C at $q/A = 25 \text{ kw/m}^2$ and the boiling curve is shifted to the left indicating high heat transfer. This may be explained in such a manner that at or near the horizontal case, as soon as bubbles depart from the surface, liquid floods into the interconnecting channels and large artificial pores and hence porous coatings are not superheated to activate nucleation sites. On the other hand, as the inclination angle increases, especially at the range of $90^{\circ} \sim 180^{\circ}$, bubbles keep the liquid away so that those artificial pores and channels are not snuffed off by the liquid. Therefore, porous coatings are superheated to activate nucleation sites, which causes a sudden superheat drop as shown in Fig. 5.14. Once these porous coatings are superheated, even though the surface orientation is changed to the horizontal case, the surface shows still far decreased

superheat, which marks another type of hysteresis effect different from the one mentioned earlier in section 5.2.

5.4 Film Boiling

Film boiling tests were conducted for three surfaces (copper, UNB #1, and UNB #2) at 2 bar as shown in Figs. 5.1, 5.3 and 5.4 respectively. The heat flux range related to film boiling, $18 \sim 90 \text{ kw/m}^2$ is relatively low as compared to that in nucleate boiling. Film boiling data could have been taken at a higher heat flux range but only few points including the $(q/A)_{\min}$ were taken because the present study was focused on the determination of the $(q/A)_{\min}$ and the corresponding $(\Delta T)_{\min}$ at different surface orientations. A fouling effect on the heating surface can be produced at high heat flux in film boiling [34] and it was wished to avoid this type of surface contamination.

In general, the film boiling coefficients are even lower than that of natural convection if the natural convection curves are extended up to the film boiling heat flux range.

There are large variations in the minimum superheat required to sustain film boiling, $\Delta T_{\min}$ with various surfaces as shown in Fig. 5.19. UNB #2 has the biggest value (152°C), next UNB #1 (123°C) and copper has the smallest value (73°C) at pressure 2 bar. This fact is very much contradictory to Berenson's [35] experimental

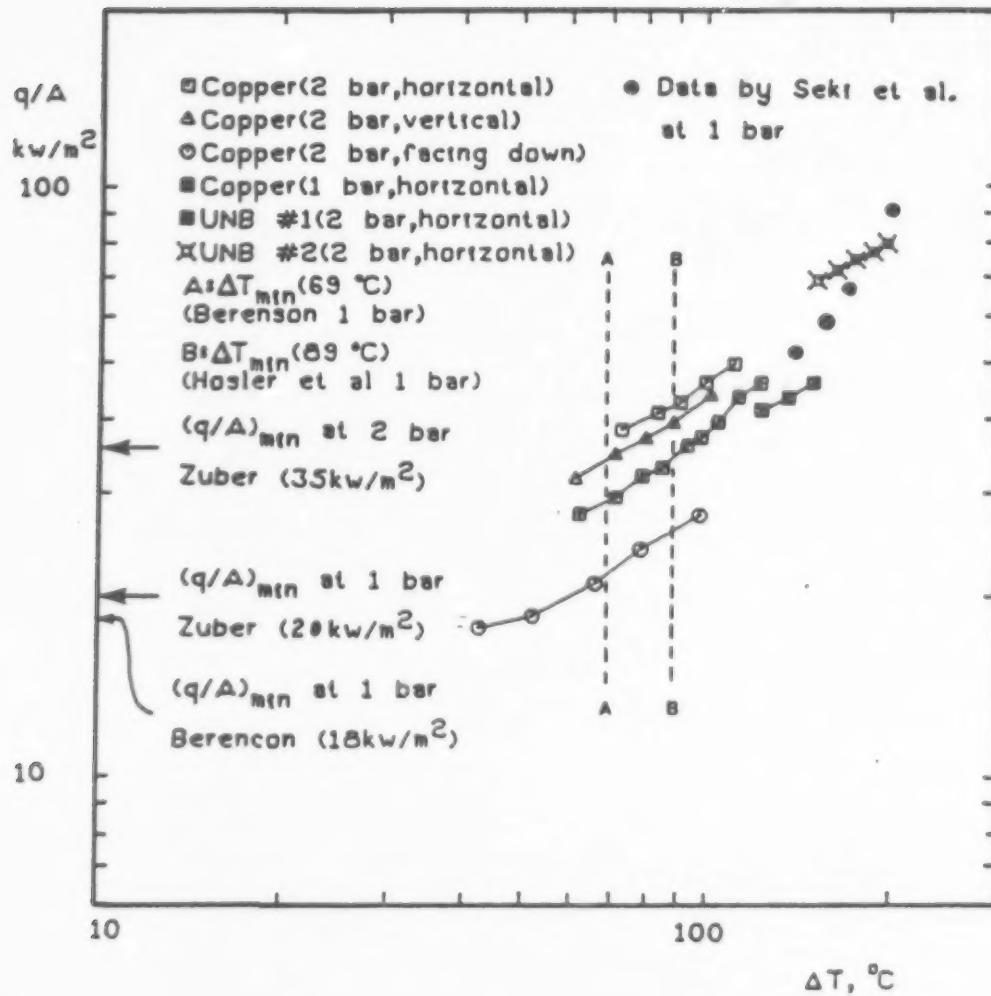


Fig.5.19 Comparison of the present results with available data and predictions in film boiling region.

results which claimed that the surface roughness does not have an effect on the $\Delta T_{\min}$.

It is very clearly seen in Fig. 5.19 and Table 5.3 that the rougher the surface is, the larger the superheat. This discrepancy may be explained by the fact that all Berenson's surfaces are very much finer than the UNB manufactured surfaces and they do not have the artificial pores.

A simple 1-D heat flow analysis can verify that the surface roughness is an important parameter in determining $\Delta T_{\min}$ in film boiling. For instance, one can assume that for facing downwards, $\Delta T_{\min}$ is composed of two portions as illustrated in Fig. 5.20. One is ΔT_{hyd} which is determined by the hydrodynamics of the fluid and assumed to be constant regardless of the roughness of the surface. The other is ΔT_{Ra} which is the superheat drop across the minimum film thickness which is assumed to be equivalent to Ra, the arithmetic mean of the departure of the surface profile from the mean line.

The analysis is very straightforward. Firstly, ΔT_{Ra} for each surface is determined by the following equation:

$$(q/A)_{\min} = K_v \frac{\Delta T_{\text{Ra}}}{\Delta x} \quad (5.2)$$

where $(q/A)_{\min}$, K_v and Δx are the minimum heat flux determined experimentally, thermal conductivity of the vapor film, and the assumed minimum film thickness, Ra,

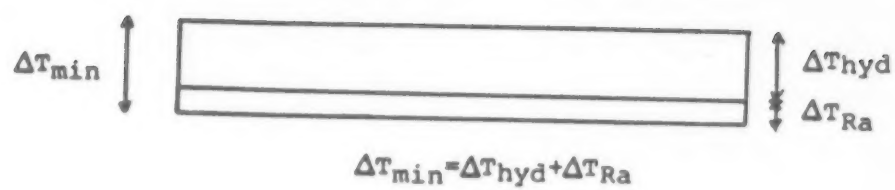


Fig.5.20 Two components of the minimum superheat, $\Delta T_{\min}$.

respectively. Secondly, ΔT_{hyd} can be calculated by subtracting the ΔT_{Ra} from ΔT_{min} which is determined experimentally for the copper surface. Thirdly, the estimated superheat drops, ΔT_{est} across the vapor films in UNB #1 and #2 can be determined by adding their own ΔT_{Ra} values to ΔT_{hyd} which is assumed to be constant as mentioned before. Table 5.4 illustrates the comparison of the Ra value, $(q/A)_{\text{min}}$, ΔT_{min} and ΔT_{Ra} for three surfaces.

ΔT_{hyd} is determined to be 39.34°C . The estimated superheat drops, ΔT_{est} for UNB #1 and UNB #2 are 51°C and 73.74 respectively, which represents 81% and 82% of the actual ΔT_{min} values for two surfaces. Out of this analysis, it is verified that the rougher the surface is, the higher the superheat.

The present results in film boiling were also compared with the predictions and experimental results reported by other workers as seen in Fig. 5.19. Zuber's [36] and Berenson's [37] equations for $(q/A)_{\text{min}}$ underpredict the actual value with the error of 15 ~ 20% (Zuber) and 36% (Berenson). But ΔT_{min} predicted by Berenson, 69°C is very close to the experimental data, 63°C at the pressure 1 bar. Hosler and Westwater's [18] experimental results show higher ΔT_{min} (89°C) because their apparatus did not permit operation at ΔT of 70°C and they believed ΔT_{min} to

Table 5.4. Comparison of the R_a , $(q/A)_{\min}$, $\Delta T_{\min}$ and ΔT_{Ra} for three surfaces at 2 bar, facing downwards

	Copper	UNB #1	UNB #2
R_a (μm)	1.56	6.61	11.7
$(q/A)_{\min}$ (kw/m^2)	17.7	15.0	24.9
$\Delta T_{\min}$ ($^{\circ}\text{C}$)	42.6	63.0	89.4
ΔT_{Ra} ($^{\circ}\text{C}$)	3.26	11.66	34.4

be lower than their value. Seki et al., either, [19] did not obtain the actual $\Delta T_{\min}$, whereas, they simply extrapolated the nucleate and film boiling curves. However, their curve for the horizontal case at 1 bar coincides with that of the present study if their curve is drawn at a lower heat flux. Out of these results it is thought that the $\Delta T_{\min}$ at 1 bar in Freon-11 would be at the order of $63^{\circ}\text{C} \pm 5^{\circ}\text{C}$.

The effect of orientation on film boiling has been observed also. As shown in Fig. 5.19, the heat transfer coefficient at the horizontal position records the highest value, which is contrary to the expectations, considering the surface orientation around 165° has the highest value in nucleate boiling. This may be explained by the fact that heat transfer mechanisms are quite different from each other.

In nucleate boiling, no vapor films are formed so that liquid access to the surface is quite easy, and what is more, the surface orientation causes the high convection currents. On the other hand, in film boiling vapor films are formed on the surface which retards the heat transfer. In the horizontal case, the vapor film is unstable because of easy liquid penetration compared to other orientations. When facing downward, the whole surface is covered with a vapor pocket (Fig. 5.21), which cuts liquid flow completely to the surface causing the poorest heat



Fig.5.21 Film boiling at facing downward.

transfer. Photographic studies reveal that the vapor motion is a strong function of the surface orientation such that for facing downward, vapor flows out in a radial direction along the heated surface, while the vapor escapes very easily to the bulk liquid for any other orientation.

As seen in Fig. 5.19, $\Delta T_{\min}$ is varied according to the surface orientation in such a manner that as the inclination angle increases the minimum temperature differences for the vertical and facing downward (61.3°C , 42.6°C) decrease at the order of 84% and 58% of the value for the horizontal case (73°C).

For film boiling it has been observed that there is a temperature gradient radially on the surface in such a manner that the center region attains the highest value which then decreases toward the edge. The order of temperature difference between the center and edge is $5 \sim 6^{\circ}\text{C}$. This means that liquid can have better access at the edges than at the center as expected.

5.2.5 Effect of Explosafe

Explosafe, which is a honeycomb type expanded aluminum mesh, has been proposed for use as an internal protection device in case of accident in railway tank cars. It was supposed that Explosafe would act as a fin to take the heat away from the point of application. The results,

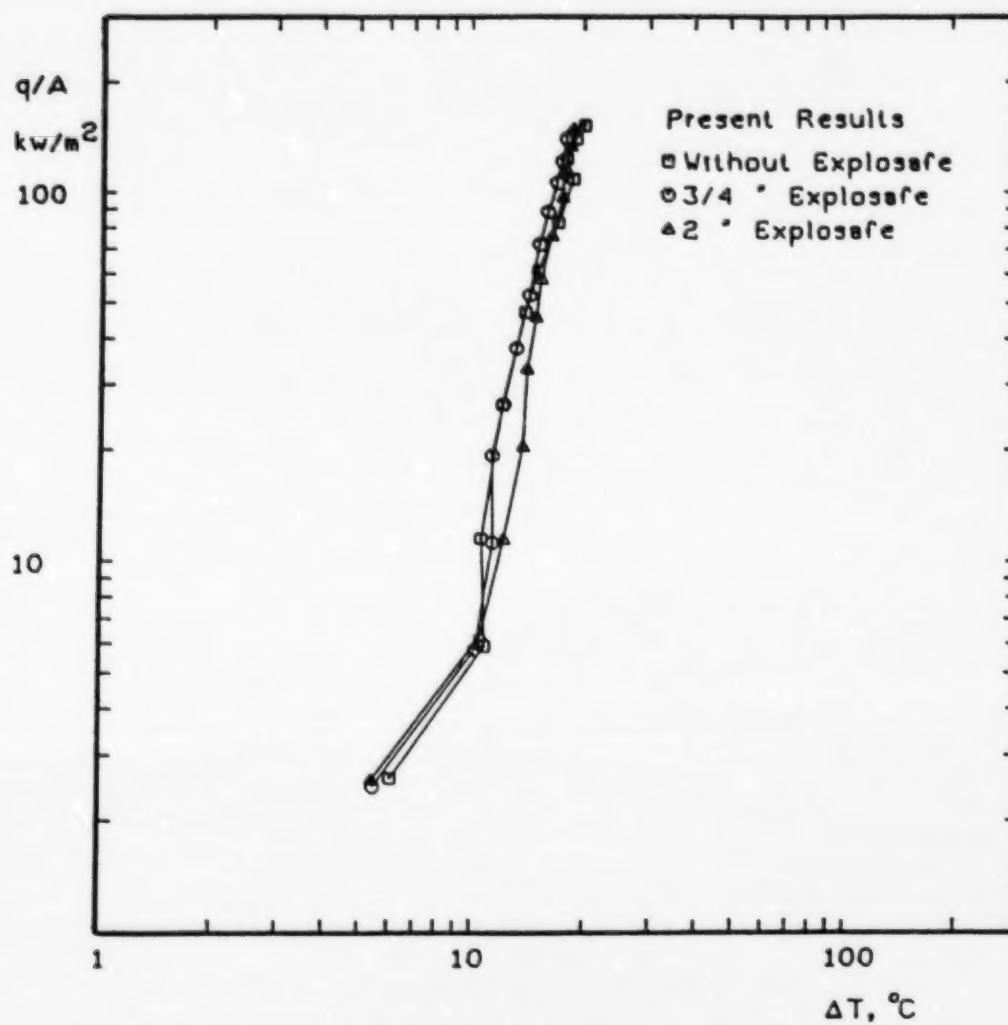


Fig.S.22 Effect of Explosafe.

however, reveal that Explosafe does not cause any appreciable enhancement in nucleate boiling. As the thickness increases, even the heat transfer was retarded (Fig. 5.22). This can be explained by introducing thermal contact resistance and bubble nucleation theories. Even though Explosafe is in good contact with the surface, there is a thin layer of liquid between the Explosafe and the surface, which cuts the heat flow toward Explosafe and subsequently, Explosafe is not superheated to activate nucleation sites out of its surface. Hence, this aluminum layer reduces to nothing but a thermal resistance to prevent the liquid from having a good access to the surface, wetting the surface and sweeping the heat away as a result. As the thickness increases, thermal resistance of the Explosafe increases causing poorer heat transfer to the liquid.

5.2.6 Effect of Pressure

As seen in Fig. 5.23, the heat transfer coefficient increases all over the boiling regimes including film boiling according to the pressure increase inside the pool. This is explained by the equation (5.3) [38] as follows:

$$T_v - T_{sat} = \frac{2R_v T_{sat}^2 \sigma}{h_{fg} P_\ell r} \quad (5.3)$$

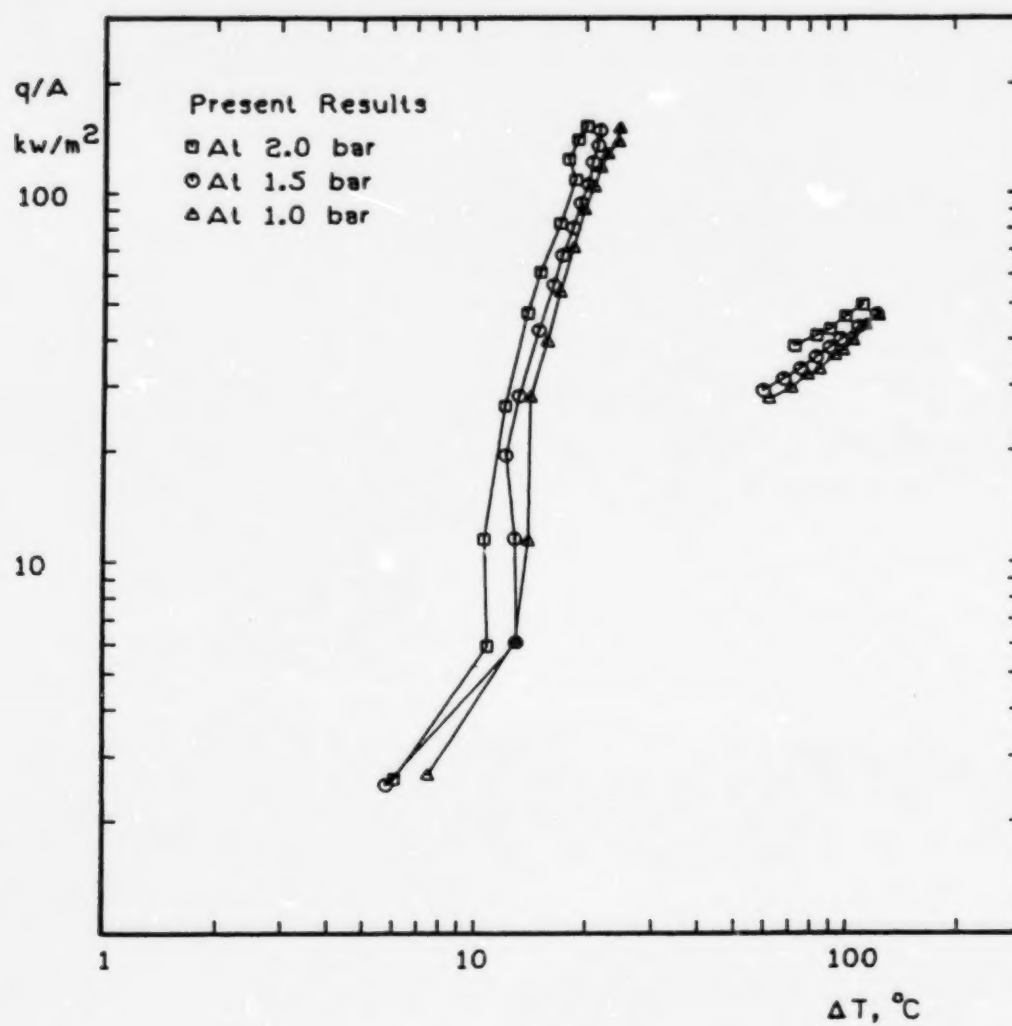


Fig.5.23 Effect of pressure.

where σ , R_v and r are surface tension, gas constant of the vapour and radius of the bubble. A particular surface has a given cavity size distribution and at a particular (q/A) a definite number of active sites are required; then the superheat required must activate a particular size of cavity of radius r . The right hand side of equation (5.3) with r constant decreases with increasing pressure, hence, less superheat is required, showing that the boiling curves for high pressure should be located to the left of those for low pressure as seen in Fig. 23.

5.2.7 Evaluation of Constants in Rohsenow Correlation and Their Variation

Recent evidence [11, 39] supports the premise that the coefficient ' C_{sf} ' and the exponent ' r ' of the Rohsenow equation vary with surface preparation technique as well as with surface-liquid combination:

$$\frac{C_{p_l} \Delta T}{h_{fg}} = C_{sf} \left(\frac{q/A}{\mu_l h_{fg}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right)^r \left(\frac{C_l \mu_l}{k_l} \right)^s \quad (5.4)$$

The uniqueness of the equation is that only one set of experimental values of q/A and ΔT is necessary to determine the coefficient ' C_{sf} ' for given ' r ' and ' s ' values. Once the coefficient ' C_{sf} ' and the exponent ' r ' are obtained, the behaviour of a given liquid-surface combination can be determined for other values of ΔT . Rohsenow originally

used values 0.011 and 0.33 for ' C_{sf} ' and ' r ' respectively but he indicated that as a minimum the coefficient and even the exponent must change in magnitude as the character of the solid surface changes. Rohsenow, however, did not mention that those constants could vary with the surface orientation.

As seen in Figs. 5.1 ~ 5.4, it is very natural that those constants should change to compensate for the differences in boiling curve caused by the surface orientation and preparations. As seen in Figs. 5.24 and 5.25, for the plain surfaces, as the surface orientation varies from horizontal to 165° , the exponent ' r ' increases slightly and the constant ' C_{sf} ' decreases very little and these two values jump to higher ones near 180° . Compared to the values obtained by Chen [11], the present values of ' r ' and ' C_{sf} ' are quite compatible with Chen's values within the deviation of 10 ~ 15%. One thing to be noted is that while in Chen's results, ' r ' and ' C_{sf} ' jump at about 150° , they jump at 165° in the present results. Considering the fact that Chen did not perform the experiments in detail at the angle of $160 \sim 180$, it is understood that the present results reflect the actual fact.

Regarding the enhanced surfaces, usually they show two times higher values than those of plain surfaces in terms of exponent ' r ' but to the contrary, they reveal 2 ~ 4

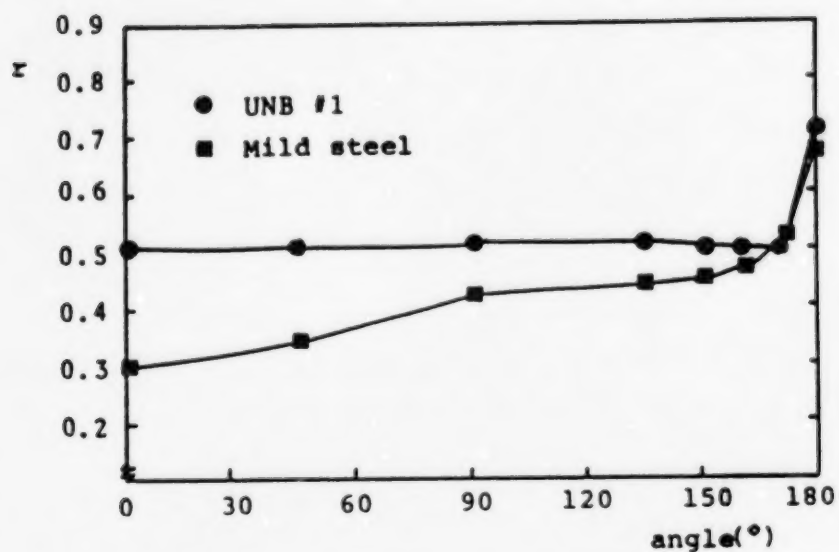


Fig.5.24 Variation of the exponent 'r' in Rohsenow equation.

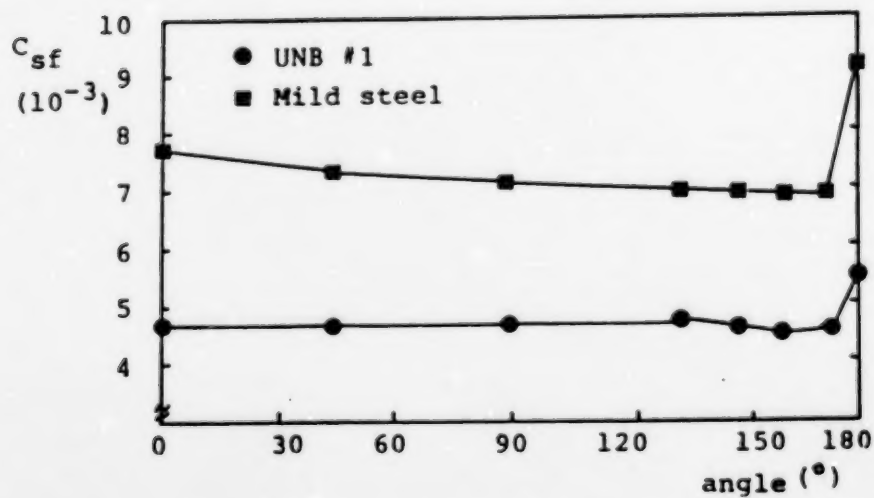


Fig.5.25 Variation of the constant 'C_{sf}' in Rohsenow equation.

times decrease in the constant ' C_{sf} ' compared to the values for plain surfaces (Tables in Figs. 5.24 ~ 5.30). Surface orientation effects, however, do not play an important role in constant ' C_{sf} ' and ' r ' as they do for ordinary surfaces. The values for ' r ' and ' C_{sf} ' keep constant all over the surface orientations except facing downward where both values again jump to record high values.

As to the effect of pressure on ' C_{sf} ' and ' r ', the present results reveal that pressure does not affect those constants because there is no appreciable change in ' C_{sf} ' and ' r ' as the system pressure changes from 1 bar to 2 bar (Fig. 5.31). The results demonstrate that pressure is not a ruling factor in determining ' C_{sf} ' and ' r ' proving that Rohsenow's equation is independent of pressure [31]. The correlation coefficient coming from the reduction of data using least square method was very good (0.96 ~ 0.98). (Note: only the data after the incipient boiling point is considered for the correlation.)

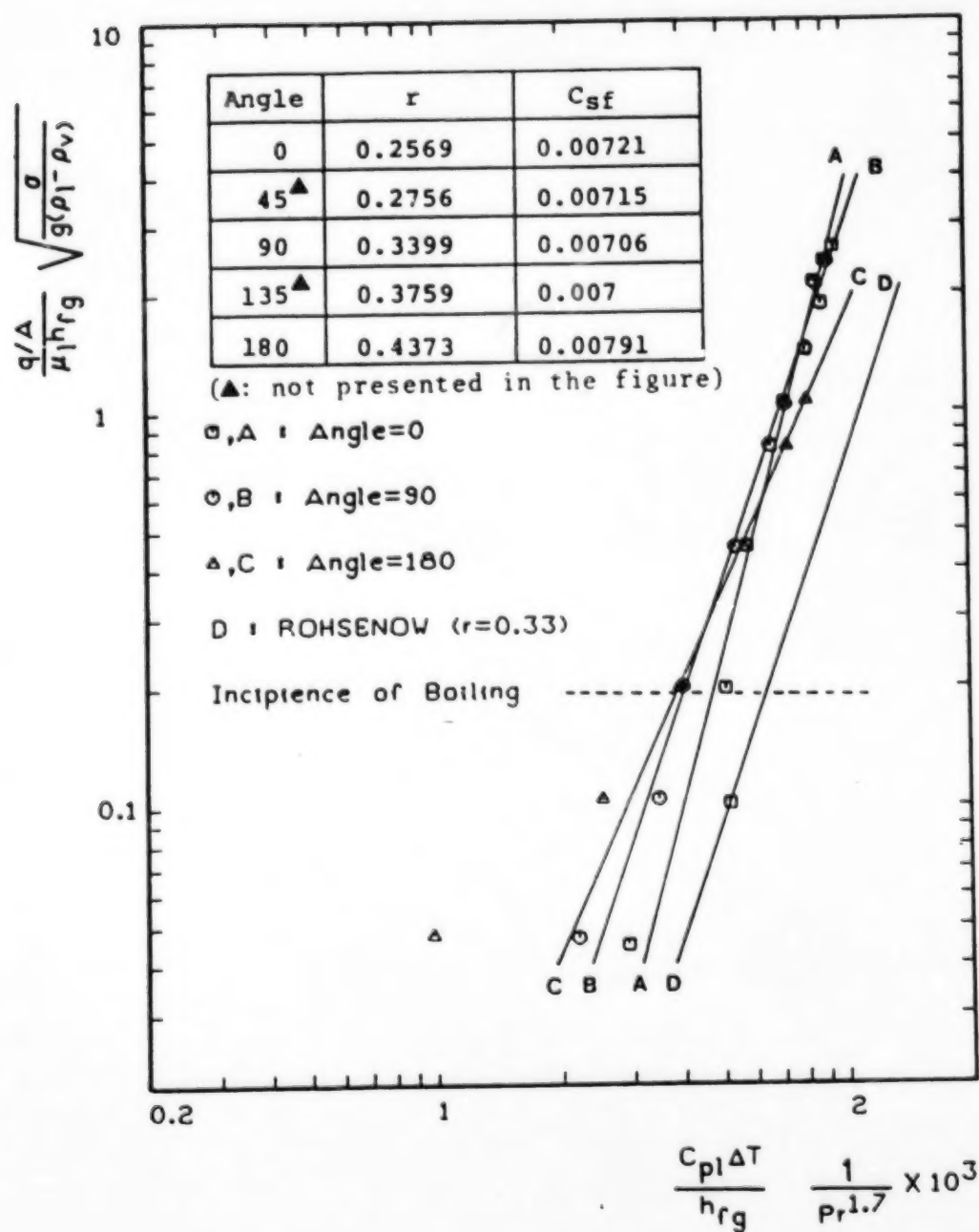


Fig.5.26 Correlation of data from copper surface using modified Rohsenow equation.

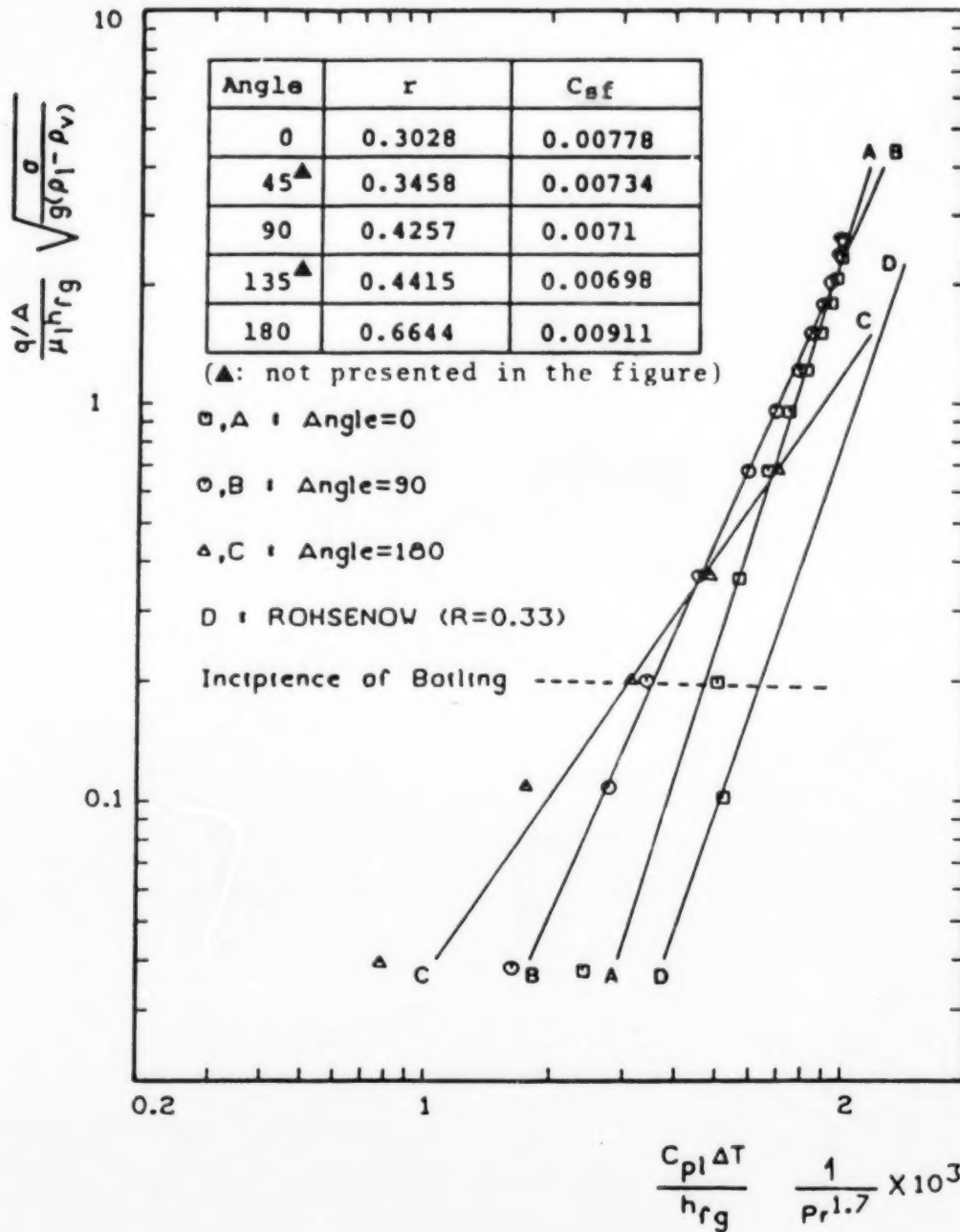


Fig.5.27 Correlation of data from mild steel surface using modified Rohsenow equation.

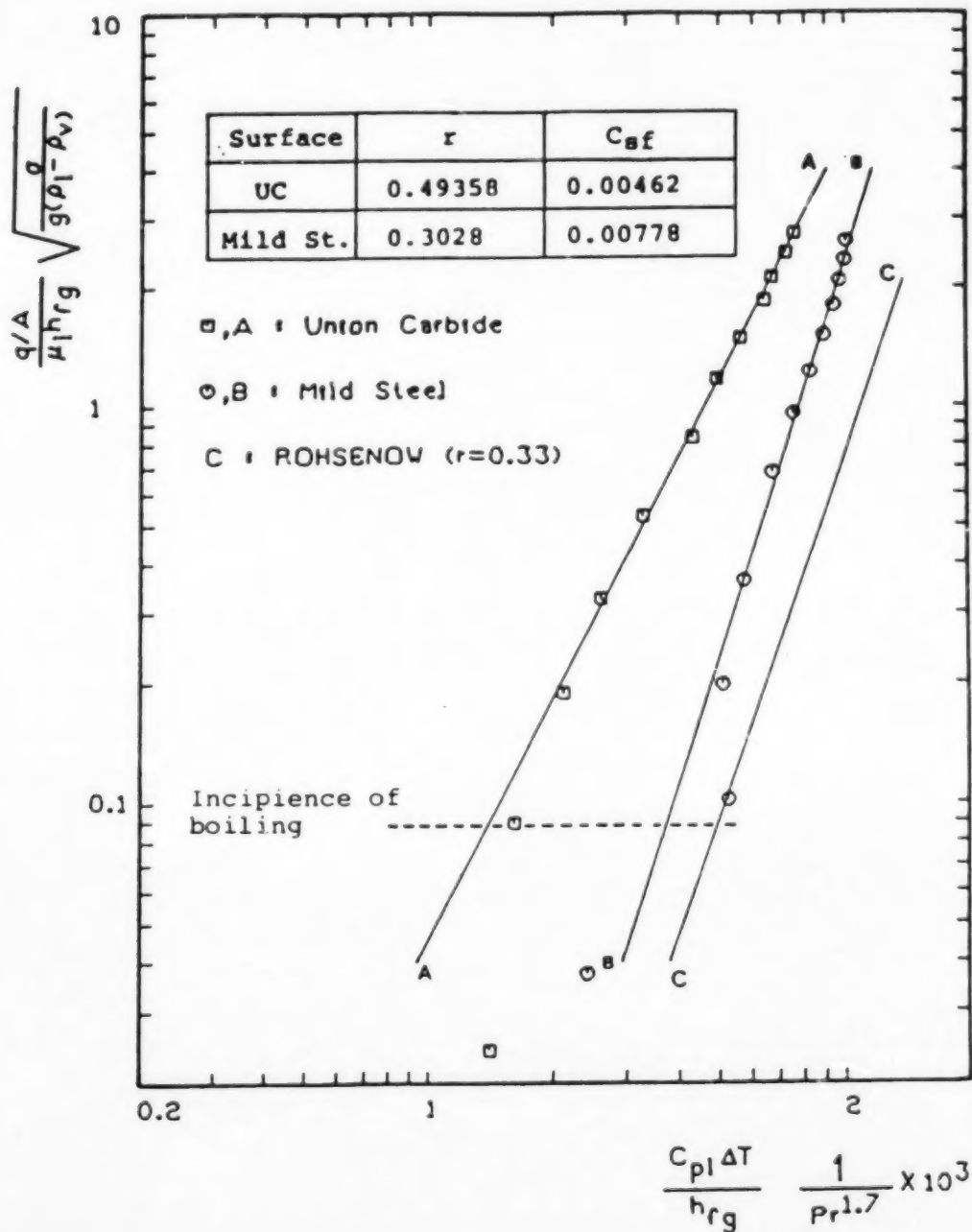


Fig.5.28 Correlation of data from UC surface, horizontal case using modified Rohsenow equation.

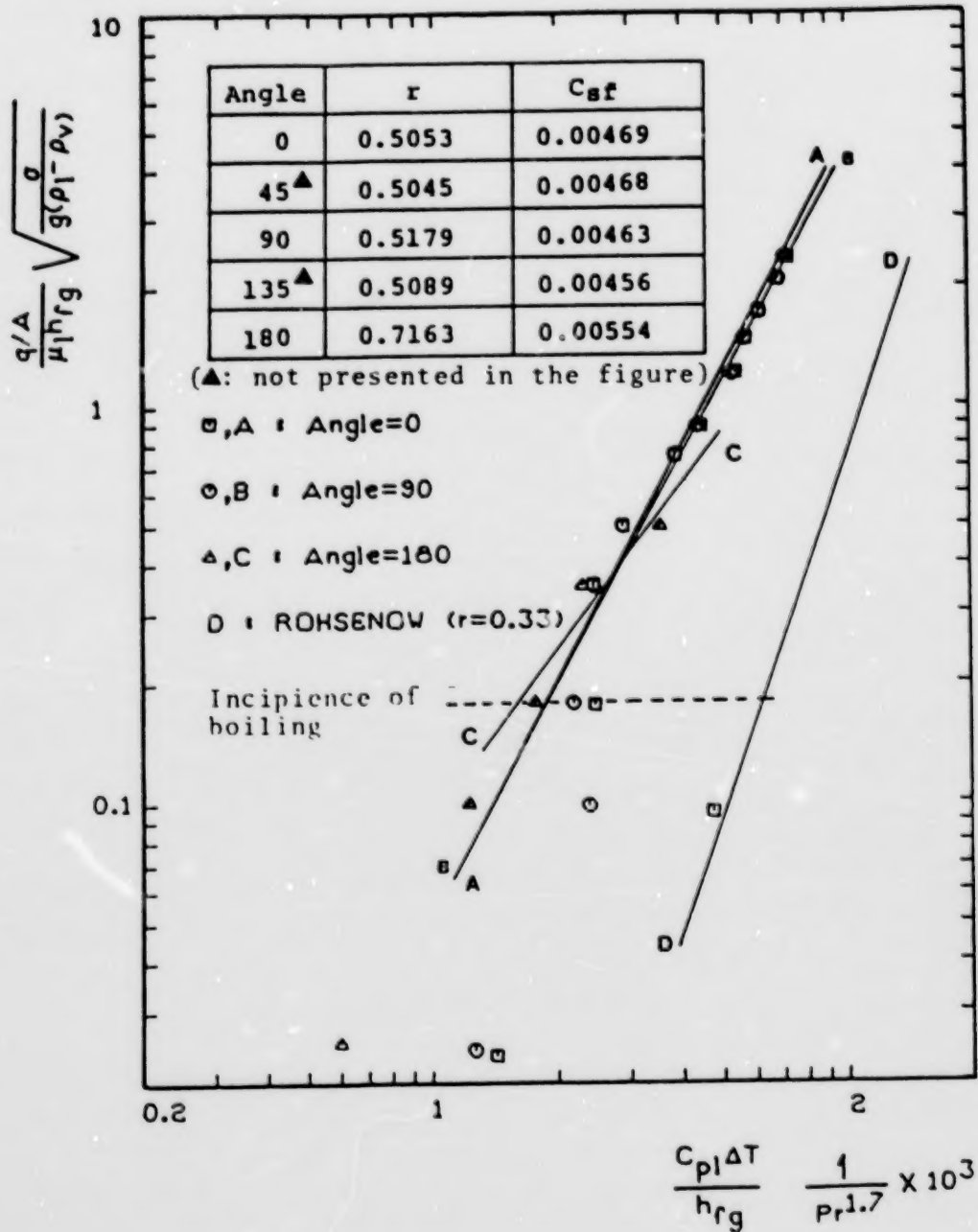


Fig.5.29 Correlation of data from UNB #1 surface using modified Rohsenow equation.

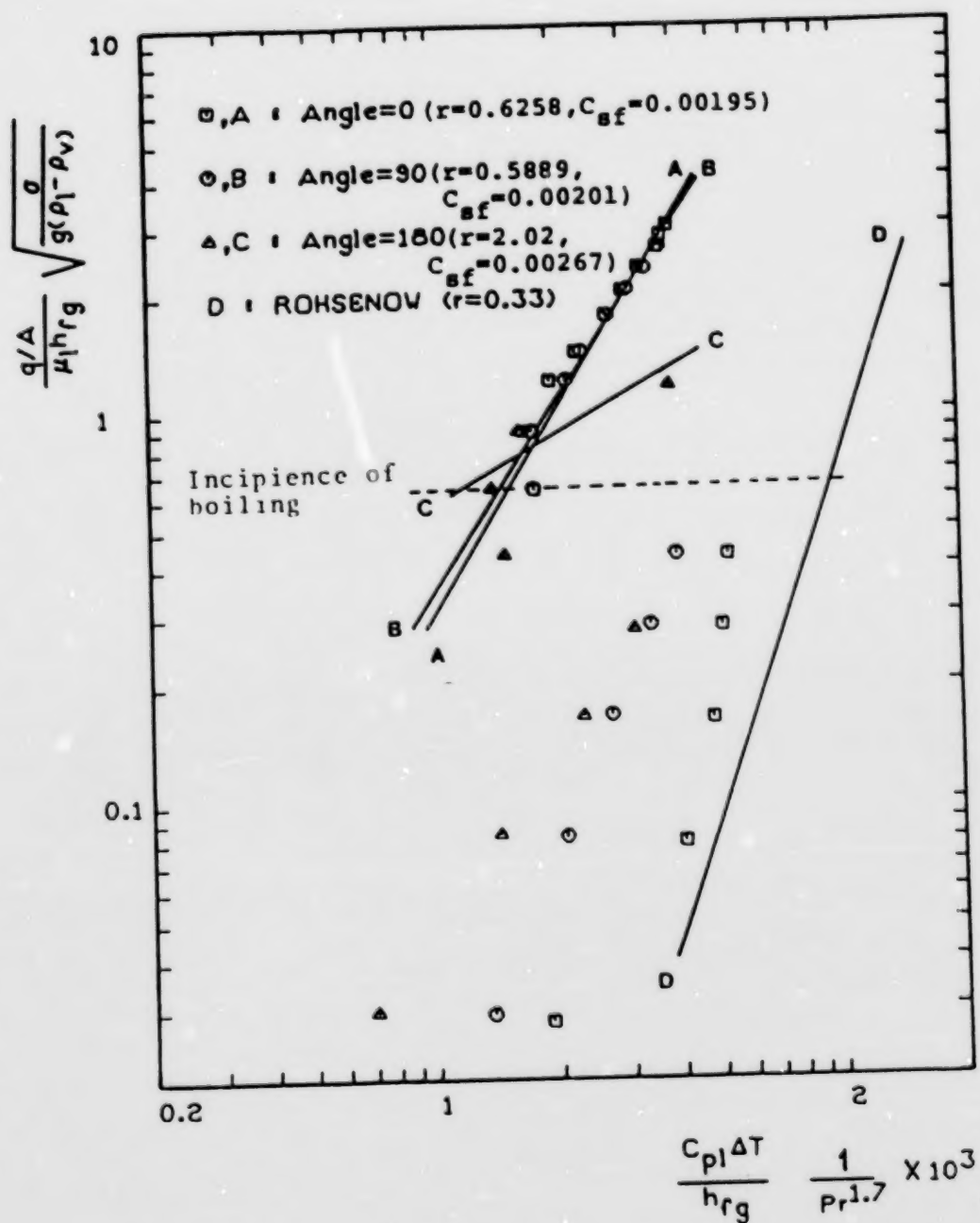


Fig.5.30 Correlation of data from UNB #2 surface using modified Rohsenow equation.

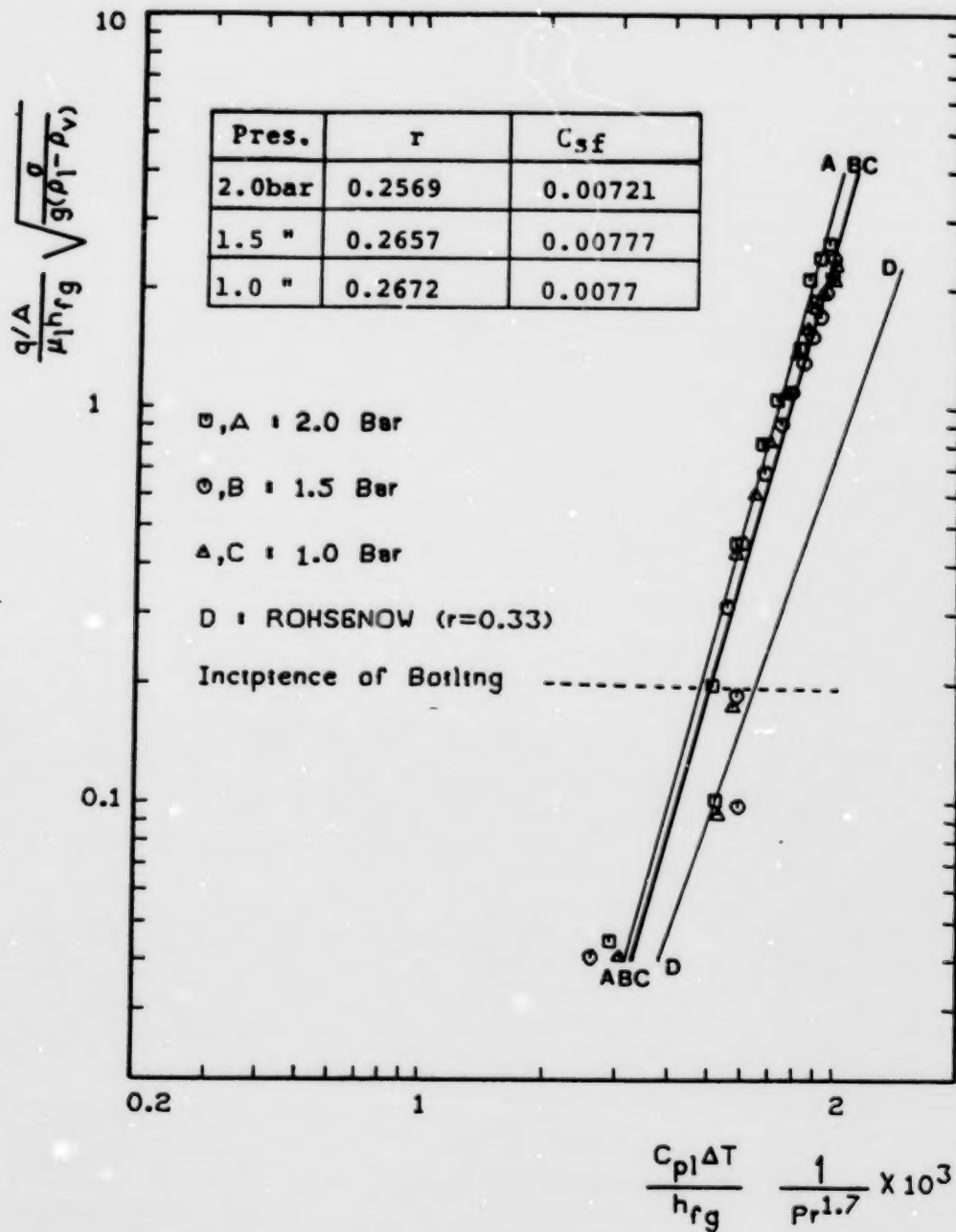


Fig.5.31 Correlation of data from copper surface at different pressures using modified Rohsenow equation.

CHAPTER 6

CONCLUSIONS AND SUGGESTIONS FOR FUTURE WORK

6.1 Conclusions

The work of Githinji and Sabersky [10], Chen [11], revealed one of the unexplored characteristics of boiling as a function of surface orientation. Bergles et al. [4] suggested the use of an enhanced surface as a method for obtaining high heat transfer. Singh [16] indicated the importance of two phase swelling in the boiling system. The experimental investigations here were carried out to develop a heat transfer enhancing device and compare their performances, to clarify the effects of orientation and swelling on boiling. Theoretically two phase swelling was analyzed. As well all data in nucleate boiling was correlated using the modified Rohsenow pool boiling correlation.

The following points have been achieved during this work:

1. Pool boiling tests in Freon-11 have been conducted on copper (with and without Explosafe), mild steel, Union Carbide High Heat Flux, and two UNB manufactured enhanced surfaces at the pressure 2 bar (Fig. 6.1). A simple experimental apparatus was designed for this test, which facilitates in taking data at any surface orientation for

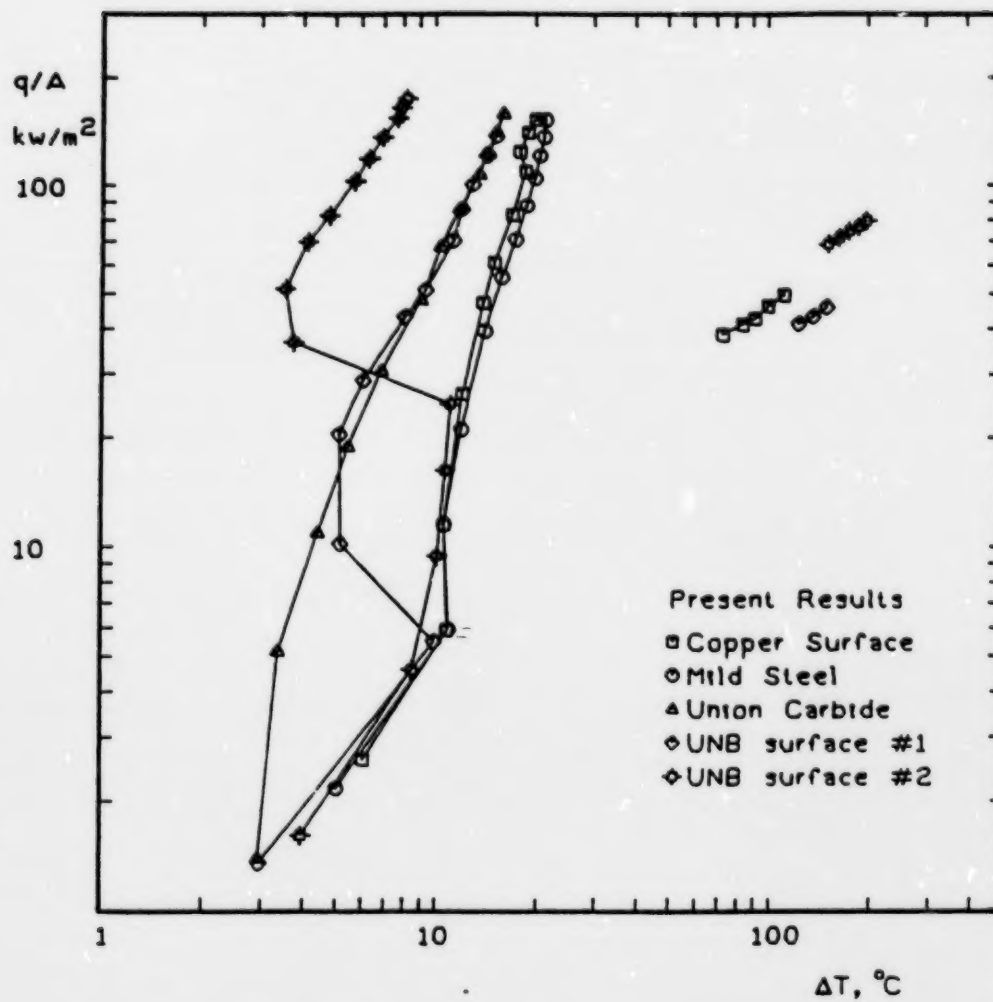


Fig.6.1 Pool boiling heat transfer with Freon-11 for various surfaces at 2 bar, horizontal case.

any regime of the boiling curve by adopting the direct surface temperature measurement technique and making use of the fully equipped HP-85 computer and data acquisition system. This simple apparatus allowed to generate film boiling without any difficulty by the consideration of a common sense that the peak heat flux for facing downward case is two or three times lower than that for the horizontal case. To investigate the effect of pressure, only the copper surface was tested at three different pressures (1.0, 1.5 and 2.0 bar). Plain surfaces with the same finish (copper and mild steel) do not show an appreciable difference but 7% higher heat transfer coefficient in copper as compared to mild steel, which eventually implies that the nature of the test surface such as thermal conductivity is not a ruling factor in boiling heat transfer.

2. UNB manufactured enhanced surfaces, UNB #1 and UNB #2 have revealed very large heat transfer improvements. It has been verified that the UNB #1 is very well compatible with the Union Carbide High Heat Flux surface which has been considered to be one of the highest heat transfer enhancing surfaces so far. However, the UNB #2 shows no improvement before its nucleation sites are initiated but once those nucleation sites act, then this surface far exceeds the UC surface performance recording 40 times increase in heat transfer coefficient for a

given superheat, ΔT as compared to plain surfaces. These achievements are attributed to high surface roughness and porosity for the UC and UNB #1 surfaces and high porosity as well as an increase in the actual heat transfer area by multi-layered coating for the UNB #2 surface.

Differences among these three surfaces are as follows:

- (a) UC has a number of small nonwetted pores due to which it shows fully developed boiling characteristics even at low heat flux.
- (b) UNB #1 has a number of big pores as compared to UC surface. This difference in pore size caused UNB #1 to go through incipient boiling with relatively higher ΔT at low heat flux because of liquid flooding into pores. Upon passing the incipient boiling point it gives the compatible performance with UC implying that UC and UNB #1 have almost the same porosity.
- (c) UNB #2 has a few pores but many interconnecting channels which provide a larger heat transfer area. Because of a shortage of small pores, it takes longer to be superheated to activate possible nucleation sites as compared to other enhanced surfaces.

3. The surface orientation effect has been determined to be critical from the inclination angle $165^{\circ} \sim 180^{\circ}$ which forms the upper portion of the cylindrical tank. In fully developed nucleate boiling regime, as for the angle 0° to 165° , this effect is not appreciable but after 165° superheat jumps to incur film boiling. This trend is extended in film boiling showing that the heat transfer performance at facing downward is the worst one.

4. The effect of swelling is dependent upon the initial filling of the tank and heat flux. This effect becomes vital especially at the initial stage of the fire engulfment of the tank which is highly filled with liquid. As the heat flux increases, the free space for the disengagement of entrapped droplets becomes smaller and smaller. Therefore, it is essential that the pressure relief valve should be designed in such a manner that it accounts for the two phase swelling.

5. In terms of the heat transfer coefficient, film boiling has the lowest value so that it should be avoided by any means. The minimum temperature difference, $\Delta T_{\min}$ to support film boiling turns out to be a strong function of the surface roughness in such a way that the rougher the surface is, the higher the $\Delta T_{\min}$. A simple 1-D heat transfer analysis has been successfully carried out to prove the effect of the surface roughness on $\Delta T_{\min}$. The surface orientation also affects the $\Delta T_{\min}$. As the inclination

angle decreases, $\Delta T_{\min}$ becomes larger.

6. Generally well known boiling equations cannot be used to correlate the data from enhanced surfaces. Rohsenow's boiling equation was used to account for the surface characteristics, surface-liquid combination, and surface orientation by modifying the constant ' C_{sf} ' and the exponent ' r '. As for the ordinary surfaces, the ' C_{sf} ' and ' r ' values vary slightly up to the angle 165° and jump to high values at 180° . Regarding the enhanced surfaces, the value for ' C_{sf} ' decreases 2 ~ 3 times, while the value for ' r ' increases 1.5 ~ 2 times and those values are almost constant up to 165° and again jump to high values above this angle. It also has been verified that the values of ' C_{sf} ' and ' r ' are not a function of pressure.

7. A computer program was developed for the boiling tests. Detailed analysis of the surface parameters by the Talysurf 5 system and photographic studies under the microscope using a polaroid positive-negative film have provided the basic ground work necessary to explain the surface characteristics which are responsible for the bubble nucleation mechanism and film boiling effects.

6.2 Suggestions for Future Work

1. If the working fluid is changed, only one experiment at the horizontal case for a given surface to determine ' r ' and ' C_{sf} ' values would be good enough to account

for the surface orientation. Since it is clear from these results that ' r ' and ' C_{sf} ' values vary with regular pattern, even if the surface liquid combination is changed, other boiling curves for different surface orientations could be drawn as long as the values of ' C_{sf} ' and ' r ' are known at the horizontal case.

2. In order to compensate for the drawback of UNB #2, it is reasonably suggested that a metal power with very small pores (UC) is sprayed on the surface and again the metal power for UNB #2 is sprayed on it. This process may activate nucleation sites in the lower part of the coating composed of nonwetable small pores resulting in the bubble formation at low heat flux just like the Union Carbide High Heat Flux Surface. As a result of this bubble generation, the upper portion of the coating, where the actual heat transfer area substantially increased due to multi-layered rough coating, is also activated easily.

3. The upper portion of the inside tank car (angle $160 \sim 180^\circ$) should be protected very well because heat transfer performance of this portion is much poorer than that of the other portion. To make things worse, this portion is filled with vapor rather than liquid in real situations and hence there are higher possibilities of tank rupture at this portion. Therefore, a fin type extended surface might be used to take into account the effect of

orientation which is vital at this portion.

4. A temperature-controlled pool boiling heat transfer apparatus is proposed to cover three boiling regimes including transition boiling. This can provide exactly the locations of $(q/A)_{\min}$, $(\Delta T)_{\min}$ in the boiling curve which are important indicators to prevent the system from film boiling. It is also suggested that a further study be carried out to find out a more proper variable instead of using Ra in determining the superheat drop across the vapor film, which will increase the accuracy of the 1-D heat transfer analysis in film boiling.

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APPENDIX A

THE MANUFACTURING PROCEDURE FOR UNB #2 SURFACE

Metal coating on the surface was done by flame spraying. This special process entails the application of metal powder to workpiece surface by means of a torch especially developed for this purpose. The torch burns a conventional oxy-acetylene mixture and the powdery. Fine mesh surfacing alloy is swept along by the force of the gas and is sprayed onto the workpiece while it melts. The procedure for the UNB surface #2 is given below. At first, clean the specimen by sandblasting. Next, set up the proper settling pressure for the oxygen and acetylene. Twenty-five psi for the oxygen and five psi for the acetylene are turned out to be good to produce normal reducing flame. Then preheat the specimen up to 300 - 400°C and spray the metal powder for 2 - 3 minutes depending upon how much thick layer is needed. Care should be taken in maintaining the uniform distance, usually 5 - 7 cms, between the flame and specimen.

APPENDIX B

DETAILED FLOW DIAGRAM OF THE SOFTWARE AND LISTINGS OF COMPUTER PROGRAM

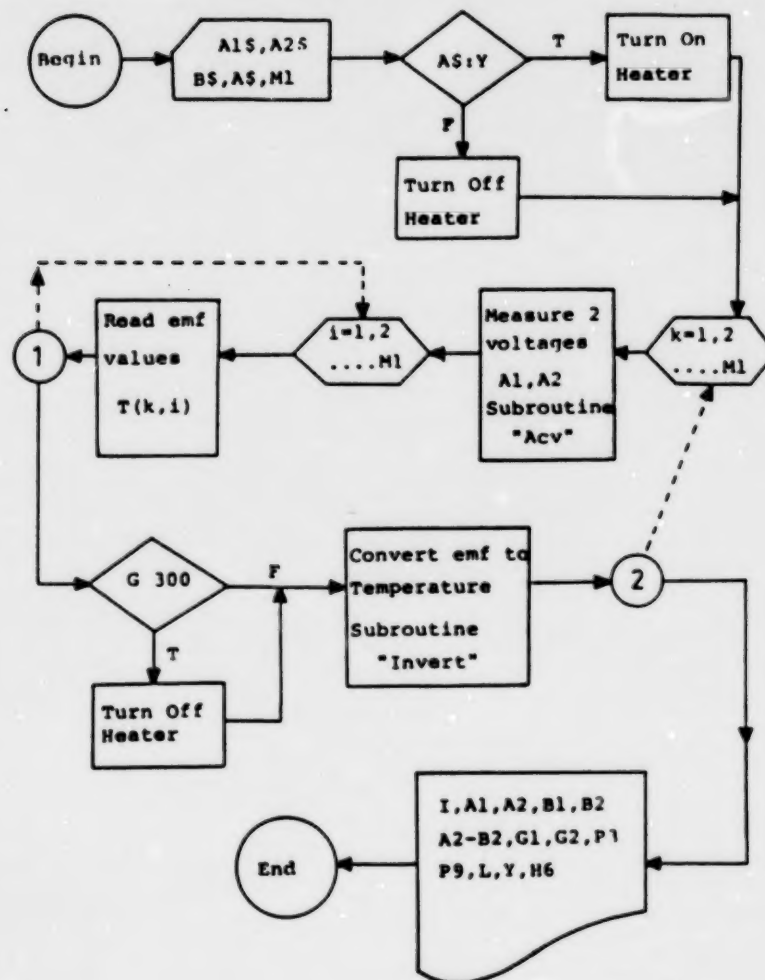


Fig.A.B.1 Detailed flow diagram for the software.

LIST OF PRINCIPAL VARIABLES

A\$	character variable to determine power on or off
B\$	character variable to determine print or display
A1\$	character variable to determine data or monitor mode
A2\$	character variable to determine fast or slow scan
A1	voltage drop across the main voltage transformer
B1	voltage drop across the standard resistor
A2	surface temperature
B2	liquid temperature
A2-B2	superheat
G	emf value difference between the present and previous scans
G1	standard deviation in surface temperature
G2	standard deviation in liquid temperature
H6	heat transfer coefficient
I	inclination angle
L	total heat loss
M1	number of scans
P3	power input
P9	percentage of total heat loss
T	matrix used for storage of emf values
Y	real heat flux
Acv	system subroutine for measurement of AC voltage
Init	subroutine for initialization of system
Invert	subroutine for inverting emf to temperature
Loss	subroutine for calculating heat loss

COMPUTER PROGRAM LISTINGS

```

1 ! THIS PROGRAM COMMANDS 3456
  DVM AND 3497A DVM TO MEASURE
  2 VOLAGES AND 11 EMF VALUE
  5.
2 ! THERMOCOUPLE NO. 1-8 ARE 1R
  ON-CONSTANTAN AND NO. 9-11 ARE
  COPPER-CONSTANTAN.
3 ! THE PROGRAM USES SOFTWARE
  COMPENSATED METHOD TO CONVER
  T EMF VALUE TO ACTUAL TEMPER
  ATURE.
4 ! TO ACHIEVE FAST CONVERGENC
  E IN CONVERTING, NEWTON-RHAP
  SON METHOD IS EMPLOYED.
5 ! SYSTEM SUBROUTINE Init AND
  ACV ARE USED FOR INITIALIZI
  NG THE SYSTEM AND MEASURE AC
  VOLAGES.
6 ! TO PREVENT HEATER FROM BUR
  NING RELAY SWITCH IS USED IN
  CASE OF SUDDEN JUMP IN SURF
  ACE TEMP.
10 OPTION BASE 1
20 DIM C1(7), C2(8), C3(6), T(60,1
  2), F(60), X1(8), Y1(8)
30 ! COEFFICIENTS FOR "J" TYPE
  AT LOWER RANGES
40 C1(1)=50.372753927
50 C1(2)=-0.030425491284
60 C1(3)=-8.5669750464E-5
70 C1(4)=1.3348825735E-7
80 C1(5)=-1.7022405966E-10
90 C1(6)=1.9416091901E-13
100 C1(7)=-9.6391844859E-17
110 ! COEFFICIENTS FOR "T" TYPE
120 C2(1)=38.74077394
130 C2(2)=-0.033190196092
140 C2(3)=2.0714183645E-4
150 C2(4)=-2.1945034823E-6
160 C2(5)=1.103190055E-8
170 C2(6)=-3.0927581898E-11
180 C2(7)=4.5653337165E-14
190 C2(8)=-2.761687804E-17
200 ! COEFFICIENTS FOR "J" TYPE
  AT HIGHER RANGES
210 C3(1)=297217.51778
220 C3(2)=-1505.9632873
230 C3(3)=3.2051064215
240 C3(4)=-.003221017423
250 C3(5)=1.5949968780E-6
260 C3(6)=-3.1239801752E-10
270 DISP "MODE ?(D/M)"
280 INPUT A$
290 DISP "FAST OR SLOW ?(F/S)"
300 INPUT A$
310 DISP "PRINT OR DISP?(P/D)"
320 INPUT B$
330 DISP "NUMBER OF SCANS ?"
340 INPUT M1
350 DISP "POWER ON ?(Y/N)"
360 INPUT A$
370 IF A$="Y" THEN 380 ELSE 400
380 OUTPUT 709 ; "DO4.S."
390 GOTO 410
400 OUTPUT 709 ; "DO4.8."
410 Q=TIME
420 A1=0
430 B1=0
440 A2=0
450 B2=0
460 A3=0
470 B3=0
480 ! SCANNING FROM 68 TO 79
490 OUTPUT 709 ; "AFOSAL79"
500 IF A2$="F" THEN 550 ELSE 510
510 P1=3456
520 GOSUB 3800 ! CALL Init
530 F1=0
540 F2=0
550 FOR K=1 TO M1
560 IF A1$="D" OF A2$="F" THEN 5
  70 ELSE 520
570 DISP "SCAN # =" ; K
580 IF A2$="F" THEN 670 ELSE 590
590 P1=30
600 GOSUB 4610 ! CALL ACV
610 A1=A1+P
620 P5=P
630 P1=32
640 GOSUB 4610 ! CALL ACV
650 B1=B1+P
660 P6=P
670 FOR J=1 TO 12
680 OUTPUT 709 ; "ASUT3"
690 ENTER 709 ; T(K,J)
700 NEXT J
710 F(K)=T(K,1)
720 IF K>1 THEN 730 ELSE 810
730 G=(T(K,1)-F(K-1))/1000000
740 IF G>300 THEN 750 ELSE 810
750 OUTPUT 709 ; "DO4.8."
760 DISP "FROM NOW ON, POWER IS
  OFF"
770 PRINT "FROM NOW ON, POWER IS
  OFF"
780 IF A2$="F" THEN 810 ELSE 790
790 PRINT "AVER. VOLT 1 =" ; P5
800 PRINT "AVER. VOLT 2 =" ; P6
810 IF A1$="D" OR A2$="F" THEN 9
  10 ELSE 820
820 GOSUB 1910
830 A2=A2+T1
840 B2=B2+T2
850 A3=A3+T5
860 B3=B3+T6
870 F1=F1+T1*T1
880 F2=F2+T2*T2
890 IF T2>46.5 THEN 900 ELSE 910
900 OUTPUT 709 ; "DO4.8."
910 NEXT K

```

```

920 DISP "DATA IS TAKEN."
930 FOR I=1 TO 20
940 BEEP I*RND+1.100
950 NEXT I
960 IF A2$="F" THEN 970 ELSE 121
970 DISP "STORE DATA ?(Y/N)"
980 INPUT A3$
990 IF A3$="Y" THEN 1000 ELSE 10
1000 DISP "ENTER DATA FILE(NAME)"
1010 INPUT C$
1020 A7=60*12*8
1030 CREATE C$,1,A7
1040 ASSIGN# 1 TO C$
1050 PRINT# 1 ; T(,)
1060 ASSIGN# 1 TO *
1070 DISP "CONVERT NOW ?(Y/N)"
1080 INPUT A4$
1090 IF A4$="Y" THEN 1120 ELSE 1
1100 DISP "NO CONVERSION NOW"
1110 GOTO 1790
1120 DISP "VOLT 1 ="
1130 INPUT J1
1140 DISP "VOLT2 ="
1150 INPUT J2
1160 P3=(J1-.2062)*(J2-.0224)*10
1170 FOR K=1 TO M1
1180 GOSUB 1810
1190 NEXT K
1200 GOTO 1790
1210 IF A1$="D" THEN 1220 ELSE 1
1220 F1=0
1230 F2=0
1240 FOR K=1 TO M1
1250 GOSUB 1810
1260 A2=A2+T1
1270 B2=B2+T2
1280 A3=A3+T5
1290 B3=B3+T6
1300 F1=F1+T1*T1
1310 F2=F2+T2*T2
1320 NEXT K
1330 PRINT "*****"
1340 DISP "WHAT IS THE DATE TODAY?"
1350 INPUT D$
1360 PRINT "TODAY IS ",D$
1370 PRINT " "
1380 DISP "INCLINATION ANGLE ?"
1390 INPUT I
1400 PRINT "INCLINATION ANGLE ="
1410 PRINT " "
1420 A2=A2/M1
1430 PRINT "SURFACE TEMP. =",A2

1440 G1=((F1-M1*A2*A2)/M1)^.5
1450 PRINT "SIGMA 1 =",G1
1460 B2=B2/M1
1470 PRINT "LIQUID TEMP. =",B2
1480 G2=((F2-M1*B2*B2)/M1)^.5
1490 PRINT "SIGMA 2 =",G2
1500 PRINT "TEMP. DIF. =",A2-B2
1510 IF A2$="F" THEN 1610 ELSE 1
1520 A1=A1/(K-1)
1530 A1=A1-.2062
1540 B1=B1/(K-1)
1550 B1=B1-.02247
1560 PRINT "AVERAGE VOLT 1 =",A1
1570 PRINT "AVERAGE VOLT 2 =",B1
1580 P3=A1*B1*2*5
1590 PRINT "POWER INPUT =",P3
1600 Q3=P3/(3.141592*.039*.039)
1610 A3=A3/M1
1620 B3=B3/M1
1630 PRINT "LAVA TEMP. 1 =",A3
1640 PRINT "LAVA TEMP. 2 =",B3
1650 PRINT "STEEL TEMP. 1 =",T7
1660 PRINT "STEEL TEMP. 2 =",T8
1670 D2=ABS(A3-B3)
1680 L1=1.4*02/.005
1690 IF A2$="F" THEN 1740 ELSE 1
1700 PRINT "INPUT HEAT FLUX Q/A"
1710 Q4=Q3-L1
1720 P7=L1/Q3*100
1730 GOSUB 3130
1740 Q1=TIME
1750 Q1=Q1-Q
1760 Q1=Q1/60
1770 PRINT "ELAPSED TIME(MIN.)="
1780 PRINT "*****"
1790 DISP "FINE"
1800 END
1810 ! SUBROUTINE INVERT
1820 R=T(K,12)*10
1830 ! SUB CONVERT R TO MICV FOR
"J" TYPE
1840 S1=0
1850 FOR I=7 TO 1 STEP -1
1860 S1=(S1+C1(I))*R
1870 NEXT I
1880 R7=S1
1890 ! SUB CONVERT R TO MICV FOR
"T" TYPE
1900 S1=0
1910 FOR I=8 TO 1 STEP -1
1920 S1=(S1+C2(I))*R
1930 NEXT I
1940 R2=S1
1950 FOR N=1 TO 11
1960 IF N=11 THEN 2100

```

```

1970 ! CONVERSION FOR "J" TYPE
1980 IF N<=8 THEN 1990 ELSE 2050
1990 S=T(K,N)*1000000+R7
2000 GOSUB 2340
2010 T(K,N)=S
2020 IF T(K,1)>200 THEN 2030 ELSE
    E 2090
2030 OUTPUT 709 ; "D04.8."
2040 GOTO 2090
2050 ! CONVERSION FOR "T" TYPE
2060 S=T(K,N)*1000000+R2
2070 GOSUB 2870
2080 T(K,N)=S
2090 DISP N;T(K,N)
2100 NEXT N
2110 DISP " "
2120 T1=(T(K,1)+T(K,2)+T(K,3)+T(K,4))/4
2130 T2=(T(K,9)+T(K,10))/2
2140 T5=T(K,5)
2150 T6=T(K,6)
2160 T7=T(K,7)
2170 T8=T(K,8)
2180 IF A2$="F" THEN GOSUB 3720
    ELSE 2200
2190 GOTO 2330
2200 IF B$="P" THEN 2210 ELSE 2280
2210 PRINT "SCAN NUMBER ";K
2220 PRINT "SURFACE TEMP.";T1
2230 PRINT "LIQUID TEMP.";T2
2240 PRINT "T5-T6=";T5-T6
2250 DISP "LIQUID TEMP.";T2
2260 DISP "TEMP. DIFF. IN LAVA";
    T5-T6
2270 GOTO 2320
2280 DISP "SCAN NUMBER ";K
2290 DISP "SURFACE TEMP.";T1
2300 DISP "LIQUID TEMP.";T2
2310 DISP "T5-T6=";T5-T6
2320 PRINT ;
2330 RETURN
2340 ! INVERT EMF TO ACTUAL TEMP
    ERATURE FOR "J" TYPE
2350 IF S=42921.9570157 THEN 26
    20 ELSE 2360
2360 ! CONVERSION AT LOW RANGES
2370 T1=60
2380 W=0
2390 S1=0
2400 FOR I=7 TO 1 STEP -1
2410 S1=(S1+C1(I))*T1
2420 NEXT I
2430 S1=S1-S
2440 S2=0
2450 FOR I=7 TO 2 STEP -1
2460 S2=(S2+I*C1(I))*T1
2470 NEXT I
2480 S2=S2+C1(1)
2490 N1=T1-S1/S2

```

```

2500 E1=ABS(N1-T1)
2510 T1=N1
2520 IF E1<=.01 THEN 2560
2530 W=W+1
2540 IF W>50 THEN 2580
2550 GOTO 2390
2560 S=T1
2570 RETURN
2580 PRINT "NOT CONVERGE"
2590 S=T1
2600 PRINT "ABS ERROR";E1
2610 RETURN
2620 ! CONVERSION AT HIGH RANGES
2630 T1=800
2640 W=0
2650 S1=0
2660 FOR I=6 TO 1 STEP -1
2670 S1=S1*T1+C3(I)
2680 NEXT I
2690 S1=S1-S
2700 S2=0
2710 FOR I=6 TO 2 STEP -1
2720 S2=S2*T1+(I-1)*C3(I)
2730 NEXT I
2740 N1=T1-S1/S2
2750 E1=ABS(N1-T1)
2760 T1=N1
2770 IF E1<=.01 THEN 2810
2780 W=W+1
2790 IF W>50 THEN 2820
2800 GOTO 2650
2810 S=T1
2820 RETURN
2830 PRINT "NOT CONVERGE"
2840 S=T1
2850 PRINT "ABS ERROR IS ";E1
2860 RETURN
2870 ! INVERT EMF TO ACTUAL TEMP
    ERATURE FOR "T" TYPE
2880 T1=43
2890 W=0
2900 S1=0
2910 FOR I=8 TO 1 STEP -1
2920 S1=(S1+C2(I))*T1
2930 NEXT I
2940 S1=S1-S
2950 S2=0
2960 FOR I=8 TO 2 STEP -1
2970 S2=(S2+I*C2(I))*T1
2980 NEXT I
2990 S2=S2+C2(1)
3000 N1=T1-S1/S2
3010 E1=ABS(N1-T1)
3020 T1=N1
3030 IF E1<=.01 THEN 3070
3040 W=W+1
3050 IF W>50 THEN 3090
3060 GOTO 2900
3070 S=T1

```

```

3080 RETURN
3090 PRINT "NOT CONVERGE"
3100 S=T1
3110 PRINT "ABS ERROR";E1
3120 RETURN
3130 ! SUBROUTINE LOSS
3140 X1(1)=200
3150 X1(2)=300
3160 X1(3)=400
3170 X1(4)=500
3180 X1(5)=600
3190 X1(6)=700
3200 X1(7)=800
3210 X1(8)=900
3220 Y1(1)=.01809
3230 Y1(2)=.02624
3240 Y1(3)=.03797
3250 Y1(4)=.0436
3260 Y1(5)=.04953
3270 Y1(6)=.05509
3280 Y1(7)=.06028
3290 Y1(8)=.06279
3300 K1=4.6
3310 K2=1.4
3320 K3=11.5
3330 T=(A3+T7)/2+273
3340 K4=6
3350 FOR I=1 TO 8
3360 C=Y1(I)
3370 FOR J=1 TO 8
3380 IF I=J THEN 3400 ELSE 3390
3390 C=C*(T-X1(J))/(X1(I)-X1(J))
3400 NEXT J
3410 K4=K4+C
3420 NEXT I
3430 L1=P3*P7*.01
3440 IF A2$="F" THEN 3460
3450 PRINT "HEAT LOSS THROUGH BO
TOM IN WATTS =" ;L1
3460 Q5=L1/(3.141592*.039^2)
3470 T3=Q5*(.004/K2+.003/K1)+A3
3480 IF A2$="F" THEN 3500
3490 PRINT "CALCULATED TEMP =" ;
T3
3500 Q2=(T3-T7)/(.0025/K1+.0015/
K2+.001/K4+.004/K3)
3510 T6=(A3+B3)/2
3520 Q3=(T6-T8)/(.004/K2+.001/K4
+.004/K3)
3530 L2=Q2*.006*3.141592*.078
3540 L3=Q3*.011*3.141592*.078
3550 IF A2$="F" THEN 3580
3560 PRINT "HEAT LOSS 2 IN WATTS
=" ;L2
3570 PRINT "HEAT LOSS 3 IN WATTS
=" ;L3
3580 L=L1+L2+L3
3590 IF A2$="F" THEN 3610
3600 PRINT "TOTAL HEAT LOSS IN W
ATTS =" ;L
3610 P9=L/P3*100
3620 IF A2$="F" THEN 3640
3630 PRINT "PERCENTAGE OF TOTAL
HEAT LOSS =" ;P9
3640 Y=P3-L
3650 Y=Y/(3.141592*.039^2)
3660 IF A2$="F" THEN 3700
3670 PRINT "REAL HEAT FLUX =" ;Y
3680 H6=Y/(T1-T2)
3690 PRINT "HEAT TRA. COEF =" ;H
6
3700 RETURN
3710 ! SUBROUTINE
3720 Q3=P3/(3.141592*.039*.039)
3730 Q2=ABS(T5-T6)
3740 L1=1.4*Q2/.005
3750 Q4=Q3-L1
3760 P7=L1/Q3*100
3770 GOSUB 3130
3780 PRINT T1-T2;Y/1000
3790 RETURN

```

APPENDIX C

HEAT LOSS CALCULATION METHOD AND POSSIBLE ERROR

As seen in Fig. A.C.1, there are heat losses through the sides and bottom of the insulation block. The heat loss calculation method is very much straightforward that simple one-dimensional heat is assumed in both vertical and radial directions in the heating element. Heat flux through the bottom is approximated as follows:

$$(q/A)_1 = k_2 \frac{T_a - T_b}{x} = \frac{T_c - T_a}{\frac{y}{k_2} + \frac{z}{k_1}} \quad (\text{A.C.1})$$

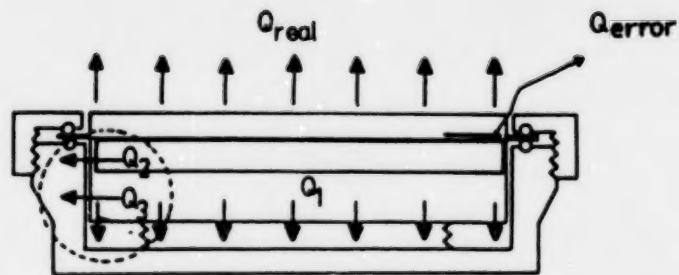
where T_a , T_b are measured values.

From eqn. (A.C.1), T_c can be calculated. Then heat flux through the upper side of the heating element is calculated by considering the thermal resistance in radial direction.

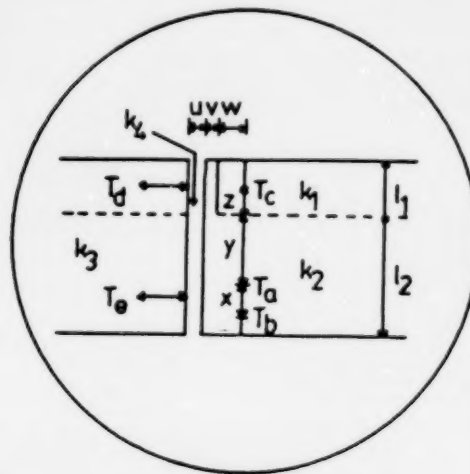
$$(q/A)_2 = \frac{T_c - T_d}{\frac{w}{k_1} + \frac{v}{k_2} + \frac{u}{k_4} + \frac{s}{k_3}} \quad (\text{A.C.2})$$

where T_d is measured value. Then, in the same manner,

$$(q/A)_3 = \frac{\frac{T_a + T_b}{2} - T_e}{\frac{(v+w)}{k_3} + \frac{u}{k_4} + \frac{t}{k_3}} \quad (\text{A.C.3})$$



(a) Heat loss through the sides and bottom of the heater.



(b) Detailed view of the circled portion in Fig.A.C.1(a).

Fig.A.C.1 Diagram for heat loss calculation.



Fig.A.C.2 Heat loss through the step of the test section.

Finally, real heat flux through the test surface becomes

$$Q_{\text{real}} = Q_{\text{gen}} - [(q/A)_1 A_1 + (q/A)_2 A_2 + (q/A)_3 A_3] \quad (\text{A.C.4})$$

where A_1 , A_2 , A_3 are areas for each section.

There is a 1 mm air gap between the heating element and stainless steel holder, where also conduction heat transfer is assumed. Goldstein and Eckert [38] pointed out that Rayleigh number of the fluids enclosed between vertical lines should be within 1000 to assume a pure conduction of the heat between the walls.

$$Ra = GrPr = \frac{\beta g (t_w - t_e) L^3}{\nu \alpha} \quad (\text{A.C.5})$$

where $(t_w - t_e)$ is the temperature difference between the heat wall and the fluid and L represents the height of the walls. As a typical example, for the walls whose temperatures are 300°C and 60°C , fluid temperature is assumed to average value of them. Then, air temperature t_e is 180°C . At this temperature, properties of air are:

$$\alpha = 0.4636 \times 10^{-4} \text{ m}^2/\text{s}, \quad \nu = 31.71 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\beta = 1/453,$$

$$L = 0.0085$$

From those values, Rayleigh number is calculated and it is 1085.59. (Note this is the maximum Ra number.) Out

of this result, the previous assumption turns out to be valid.

There may be an error in the real heat flux. As seen in Fig. A.C.1, after subtracting Q_1 , Q_2 and Q_3 from the total heat generated, Q_{real} is assumed to be conducted only vertically but there is again a heat loss, Q_{error} , through the step of the test section. As seen in Fig. A.C.2, Q_{real} is distributed two dimensionally through Area 1 and Area 2. The maximum heat through Area 1 becomes:

$$Q_{\text{error}} = Q_{\text{real}} \times \frac{\text{Area 1}}{\text{Area 1} + \text{Area 2}} = 0.0307 Q_{\text{real}}$$

So, the maximum error bound in real heat flux becomes 3.07%.

- [38] Goldstein, R.J. and Eckert, E.R.G., "The Steady and Transient Free Convection Boundary Layer on a Uniformly Heated Vertical Plate", Int. J. Heat and Mass Transfer, vol. 1, pp. 208-218, 1960.